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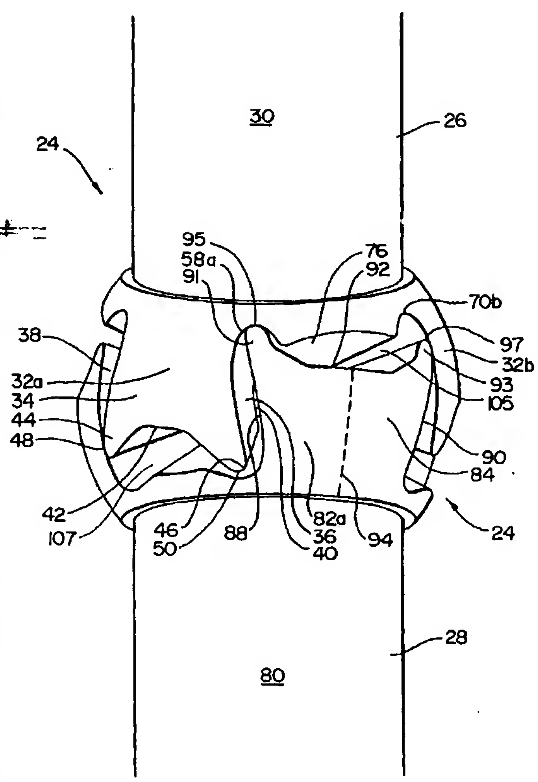
(54) Title: BALANCED ROTORS POSITIVE DISPLACEMENT ENGINE AND PUMP METHOD AND APPARATUS



(57) Abstract: A machine (20) to convert energy providing positive displacement of a fluid contained in operating chambers (105). The machine can either increase the pressure of a fluid or extract energy from a pressure differential to a rotating shaft (30). The machine having desirable balance features about various axes (10a, 10b, 12a, 12b, 14a, 14b) of the rotors (26, 28). The machine additionally having desirable axial flow characteristics to pass fluids substantially in the axial direction.

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BALANCED ROTORS POSITIVE DISPLACEMENT ENGINE AND PUMP METHOD AND APPARATUS

Related Applications

This application claims priority of U.S. Provisional Application Serial Numbers 60/218,228 which was filed July 14, 2000, 60/299,181 filed June 19, 2001 and a U.S. Provisional Application filed on July 11, 2001 (serial number not yet received).

Field of the Invention

The invention relates to positive displacement machines that convert energy, namely positive displacement devices that displace fluid (incompressible or gas) where the device has continuous rotation to displace fluid contained in operating chambers. The present invention is particularly advantageous for providing balance for the rotors about the various axes of the apparatus.

Background

This invention concerns an advanced rotary positive displacement engine having high power to mass ratio and low production cost. Engine as used in this patent document is taken to be a device that converts one form of energy into another.

In the case of prior art combustion engines, the reciprocating piston type is most widely used for its low cost of production and efficient sealing, while the turbine has shown that an external combustion engine may offer greater power partially from high speed. Rotary engines such as the Wankel engine have shown higher power to weight ratios than reciprocating engines but at the expense of increased fuel consumption. The present invention is a rotary device that offers many of the advantages of these prior art devices without many of their shortcomings.

In the case of pumps, there are many general types of pump design known, such as positive displacement, centrifugal and

impeller. Pumps of the positive displacement type are typically reciprocating or rotary.

Many previous rotary combustion engine designs have been of the single plane type in which rotary motion occurs about axes that are parallel to each other.

The present invention is of the rotary positive displacement type, but is in a class by itself. This rotary positive displacement device is believed to be the first rotary engine in which the axes of the moving parts are offset from each other and the moving parts rotate at a constant velocity relative to each other when they are rotating at a constant velocity relative to the casing. The engine is formed by a pair of facing rotors that are axially offset from another and whose faces define chambers that change volume with rotation of the rotors.

One of the limitations of certain types of fluid pumps is the inconsistent rotational force on the rotors caused by the force of the fluid pressure acting on the rotors as they rotate. In a gear pump, for example, the non-meshing portion of each of the gears is exposed to a consistent fluid pressure at the discharge port, but in the area where the gear teeth mesh together, each tooth on each rotor seals a tooth on the opposing rotor from the pressure of the fluid on the output side of the pump. This creates a rotationally imbalanced situation where each rotor alternates between balanced (with equal surface area exposed to the high pressure fluid on both "sides" of their center axis) and imbalance (with one "side exposed to the high pressure fluid, and one "side" sealed from this high pressure fluid by a tooth on the opposing rotor). The term "side" refers to one half of the total surface area exposed to the high pressure fluid of the outlet port of a pump, compressor, hydraulic motor, flow meter, actuator, or other related device.

In general, it is desirable to have the rotors of the present invention balanced about the various axes further defined herein. First, is desirable to balance slave rotor (the rotor without a torque placed thereon) about its axis of rotation. In other words, at every phase of rotation the pressures acting upon the surface area extending radially from the axis of rotation of the slave rotor cancel each other out so the slave rotor is not biased to rotate about its axis of rotation when at an operating rotational velocity. As described further herein, this balance about the slave rotor's axis of rotation is accomplished in the preferred form by providing a communication gap between the tip of the lobe of the power rotor and the base region of the slave rotor. This passage provides a conduit at the bottom dead center portion of rotation of the rotors so the fluid has access to subchambers and the net tangential forces about the slave rotor are a sum of zero. Therefore, allowing the slave rotor to be substantially unbiased throughout the various rotational positions about its axis of rotation.

A second balance issued for the rotors is hydraulic balance about the transverse axis. The transverse axis is defined an axis substantially orthogonal to the axis of rotation and extending between the regions of top dead center and bottom dead center of the rotors. In an imbalance design about the transverse axis, the high-pressure port side has a tendency to rotate the rotor about the transverse axis whereby causing the rearward region of the rotor to be biased toward the casing and in some instances causing contact between the rotor and the casing. A rotor back face having a desirable conical angle with respects to the axis of rotation can alleviate and prevent the transverse axis imbalance.

The problems which result from this hydraulic rotational imbalance are mostly related to an inconsistent rotational contact force or varying gap clearance between the moving parts. As the

parts rotate, the fluid force will act on each of the rotors to cause it to rotate forward or backward relative to the rotation of the other rotor. The inertia of the rotors themselves, the rotational velocity of the rotors, and the viscosity of the fluid, are all factors which determine at what pressure and at what speed rotors can operate without breaking through the fluid film and causing rotor to rotor contact.

Rotor to rotor contact can occur at certain operating conditions (e.g. elevated pressure or low viscosity operating fluid). Some rotor to rotor contact can be tolerated to a certain extent depending on material of the rotors and other factors, but the intermittent contact that can occur by hydraulic rotational imbalance under these conditions can cause damage or wear to the contacting parts at certain pressures and speeds and can further cause damage to sensitive fluids (e.g. blood).

The higher the fluid viscosity, the "stiffer" the fluid film, and the higher the pressure an "imbalanced pump" can tolerate without contact occurring. Speed also increases the fluid film rigidity but speed also has the detrimental effect of increasing the "impact" or “shock” characteristic of the hydraulic rotational imbalance as the pump gears (or rotors) switch back and forth from balanced to imbalanced. For certain pump configurations, it has been found that the beneficial fluid film "stiffness" effects of speed is very closely counteracted by the detrimental effects of speed due to the increased "impact" force.

To the best knowledge of the applicant, gear pumps, for example, are not used in many high pressure, low fluid viscosity applications due to the hydraulic rotational imbalance.

In the case of a pump such as the single face Outland CvR™ pump, the effect of any hydraulic rotational imbalance is even

greater due to the high volume output and corresponding high surface area which the high pressure fluid acts on.

The most significant characteristic of apparatus of the present invention is the rotational hydraulic balancing of the slave rotor. This is accomplished by allowing fluid to flow past the power rotor tips at "bottom dead center" (BDC) while having a seal at the slave rotor tips at BDC. The surface area of the slave rotor which is exposed to the high pressure is within approximately 5% -- 10% at "top dead center" (TDC) as it is at BDC at all times. This is compared to a 100% difference between top and bottom surface areas on a pump such as the Outland CvR^{TM} pump as disclosed in U.S. Patent 5,755,196.

The hydraulic rotational imbalance of the present invention is approximately 2-5% of the hydraulic rotational imbalance of the single engagement face per lobe Outland CvR^{TM} pump. This means that the pressure which could cause rotor to rotor contact with this new pump design disclosed herein (with DTE Oil Light) would be significantly greater than with a rotationally imbalance design. With thicker fluids this pressure would have to be even greater.

In addition, the remaining imbalance does not occur as the rotors enter and leave the ports, but results from the movement of the contact point around the tip of the slave rotor. This reduces the "impact" characteristic still further and allows higher fluid pressures and lower fluid viscosity without contact between the rotors.

The power rotor, with this new pump design, has a consistent torque applied thereto as a result of the fluid pressure acting upon the radially extending surface where the full surface area of each lobe is exposed to the high pressure fluid at the outlet port at TDC but not at BDC. This pressure distribution scheme is necessary for output work to be carried out by the pump (or compressor, hydraulic motor, actuator, flow meter or other related

device or function). The important characteristic of this pump is that the slave rotor "floats" rotationally and can therefor be positioned rotationally by the fluid film of low "stiffness" between it and the power rotor. Furthermore, if a fluid film does not exist due to operation conditions (drawing a vacuum from the inlet port, for example) the force between the rotors is low enough to be within the allowable "PV" value of many available materials. When an incompressible fluid is not present to establish a fluid film, it is likely that either the pressure is low enough to not create the imbalanced shock (i.e. drawing a vacuum). If there is high-pressure while a compressible fluid is present which may not establish the fluid film the presence of a compressible fluid would act as a shock absorber thereby reducing the impact effect.

With a gap clearance design between the opposing rotor's engagement tip and contoured face, it is desirable to have a fluid film gap between the engagement tip and the engagement face when they are in a location exposed to a port. This fluid film gap allows for proper rotational positioning of the rotors with respects to each another. When an adjacent engagement tip and a contoured face are located in a sealed zone (i.e. either at top dead center or the bottom dead center in a noncompressible fluid embodiment), there is a pressure difference between the minimum gap between the tip and the face and a fluid film seal is created. A seal creates a fluid film seal or a contact seal (e.g. an interference fit) with sufficient fluid resistance to prevent significant backflow of fluid from the high-pressure port to the low-pressure port.

Creating this hydraulic rotationally balanced characteristic has been accomplished in this new pump design by allowing fluid to flow past the power rotor tips at BDC. This is done by removing material from the slave rotor at the base region providing a greater fluid gap clearance between the tip of the power rotor and the base

region of the slave rotor to provide fluid communication to subchambers located at the BDC location. A seal is maintained between the rotors in this phase of rotation by adding material at the base region to the power rotor to allow it to seal against the slave rotor tip as each slave rotor tip enters the sealed zone at BDC. This seal is maintained between each slave rotor tip and the power rotor until each slave rotor tip passes from the output port to the input port at BDC.

Further, the slave rotor can also be used as the drive rotor if a consistent contact force between the rotors is desirable. This might be the case with a single direction pump.

Summary of the Invention

The invention comprises a machine that converts energy such as a pump to increase the pressure of a fluid, or a motor, turbine, flow meter or actuator taking a pressure differential in a fluid to create rotary motion about a shaft or other device that employs positive displacement of fluid (incompressible or gas). The invention comprises a housing that has an inner surface. A first rotor is mounted for rotation in the housing about a first axis and has a first outer surface that is adapted to intimately engaged the inner surface of the housing. There is further a second rotor having a forward portion and a rearward portion and is mounted for rotation and the housing about a second axis that is offset from the first axis and being collinear by an angle α and intersects at a common center of the rotors. The second rotor has a second inner surface that defines at least part of a sphere having a common center with the center of the first rotor. There is a second outer surface that is adapted to engage the inner surface of the housing. The first rotor further has a first contact face that is defined by a locus formed by points on the second rotor as the second rotor rotates about the

second axis and the first rotor further has a first contact surface positioned in the forward region of the first rotor.

The second rotor further has a second contact face that is defined by a locus formed by points on the first rotor as the first rotor rotates about the first axis. The second rotor further has a rearward surface that is positioned in the rearward portion of the second rotor. The points of each rotor that define the locus along an outer edge of a common central axis is essentially a radius extending outward from the common centers of the rotor at an angle $\alpha/2$ from the normal to the axis of the other rotor.

Where the second rotor is substantially balanced at an operating speed about various axes.

Brief Description of the Drawings

Fig. 1 is an isometric view of the apparatus and also showing an axis system used to define portions of the slave and power rotors;

Fig. 1A is a top view of a spherical master rotor on axial shaft lying on axis A at an angle α to axis B prior to modification of the rotor in accordance with the principles of the invention;

Figs. 1B and 1C are a side view and isometric view respectively of the master rotor of FIG. 1A;

Fig. 2A is a top view of a master rotor having material removed from the side of the rotor opposed to the axial shaft leaving a conical face with the apex of the cone at the center of the sphere with its axis aligned with the axis A;

Figs. 2B and 2C are a side view and isometric view respectively of the master rotor of FIG. 2A;

Fig. 3A is a top view of the master rotor of FIG. 2A with a vertically oriented cone of material conceptually overlaid on the front face of the master rotor, the cone having its apex at the

intersection of axis A and axis B (same as the center of the master rotor sphere);

Figs. 3B and 3C are a side view and isometric view respectively of the master rotor of FIG. 3A;

Fig. 4A is a top view of the master rotor of FIG. 3A showing the movement of the conceptual cone in the frame of reference of the master rotor as would be traced by the conceptual cone if it were attached to the front face of an essentially identical rotor (slave rotor) lying on axis B and having a center at the point of intersection of axis A and axis B and if the slave rotor was rotated through 180° with the master rotor from the vertical position (the conceptual cone is shown starting off center but it should be appreciated that the axis of the cone begins its movement at top dead center, corresponding to the point of lowest compression in the engine of this invention);

Figs. 4B and 4C are a side view and isometric view respectively of the master rotor of FIG. 4A;

Fig. 4D is an isometric view of the cutting, where the column has a final apical angle ψ that is a sum of $\psi1$ and $\psi2$ wear $\psi1$ is radius of the adjacent engagement tip of the opposing rotor and the value of $\psi2$ is the fluid film thickness gap and between the adjacent engagement tip and the engagement face being constructed by the conceptual cone;

Fig. 5A shows the trace of the center of the conceptual cone of FIG. 3A on the surface of the master rotor while the slave rotor and master rotor make one revolution about their respective axes;

Fig. 5B shows the trace of FIG. 5A seen in the A axis direction;

Fig. 6A is a top view of the master rotor of FIG. 4A showing an actual cone of material added to the front face of the master

rotor, the cone having its apex at the intersection of axis A and axis B, with the axis of the cone lying along the face of the master rotor whose surface is tangential to a contact face of the master rotor;

Figs. 6B is a side view of the master rotor of FIG. 6A;

Figs. 6C is an isometric view respectively of the master rotor where a second conceptual cone traces the base teardrop line as the cone and master rotors rotate about the B and A axis respectively to define an opposing engagement face;

Fig. 7A is an isometric view of the second engagement surface with the engagement tip attached to the forward region of the engagement surface;

Fig. 7B is an isometric view showing a third engagement face similar to the first engagement face;

Fig. 7C is isometric view of the second and third engagement face is with the material interposed thereinbetween removed;

Fig. 8A is a computer-aided drawing showing the axis A and axis B offset from being collinear and intersecting at a common intersect point with a reference axis bisecting the two axis on the obtuse angle portion of the axes;

Fig. 8B shows an array of the reference axis about the be axis;

Fig. 9c shows the array of reference axis rotated about the axes A to construct the base reference line;

Fig. 8D shows a base reference line offset toward the origin;

Fig. 8E shows an offset ideal surface from the base reference surface;

Fig. 8F shows a and engagement tip cone attached to the forward region of the ideal reference surface;

Fig. 9 is a front view of the rotor assembly at the bottom dead center position;

Fig. 10a – 10s shows a number of views of the rotor assembly in one position but taken from isometric perspectives in approximately twenty degree increments;

Fig. 11 is a side view of the rotor assembly showing the axis of each rotor;

Fig. 12 shows in detail the contact surface of the master rotor;

Fig. 13 is another view of one of the contact surfaces of the master rotor;

Fig. 14 illustrates the contact surface of the slave rotor;

Fig. 15 is a second close-up view of the slave rotor illustrating the gap that is provided between the same of the master rotor and the rearward surface of the slave rotor;

Fig. 16 shows the gap which is present between the master and slave rotors at the bottom dead center position to allow a pressure difference be distributed therethrough;

Figs. 17a – 17c shows three substantial front of views of each of the three slave lobes to illustrate the pressure balance acting upon these lobes;

Fig. 18a -- 18f illustrates the motion of the master and slave rotors as they pass through the bottom dead center position, whereas each progressive figure shows approximately fifteen -- twenty degrees of rotation;

Fig. 19 is a side view of a second embodiment of the present invention that allows axial flow of a fluid;

Fig. 20 is an isometric view of the contoured surfaces of the power rotor;

Fig. 21 is a rearview of the power rotor.

Fig. 22 is an isometric cross-sectional of view a fourth embodiment of the present invention;

Fig. 23 is a side view of the apparatus of the fourth embodiment showing the external surfaces of the casing portions;

Fig. 24 is a cross-sectional horizontal view of the fifth embodiment taken at line 24 – 24 in Fig. 34, and illustrating the fluid flow through two rotor sections in a in-serial arrangement;

Fig. 25 is an isometric view of a slave section of the housing;

Fig. 26a is an isometric view of the inward portion of a slave rotor;

Fig. 26b is an isometric rearview of the rearward portion of the slave rotor;

Fig. 27 is an isometric view of the inward portion of the power casing;

Fig. 28 is a rearward view of a power casing section of a casing portion;

Fig. 29 is a top view of the inward portion of the power rotor casing;

Fig. 30 is a rearview of the power rotor casing;

Fig. 31 is an isometric rearview of the power rotor of the fourth embodiment;

Fig. 32 is an isometric view of the inward portion of the power rotor of the fourth embodiment;

Fig. 33 is an isometric view of the cap employed in the fourth embodiment;

Fig. 34 is a horizontal sectional view of the fourth embodiment in an in-parallel arrangement;

Fig. 35 is an isometric view of an interior cap used in the in-parallel arrangement flow;

Fig. 36A shows an in-combination flow arrangement;

Fig. 36B shows an in-combination flow arrangement where three rotor sections are shown in an in-parallel flow arrangement

followed by an additional three parallel rotor sections positioned in-serious with the first section;

Fig. 37 is an isometric view of the slave casing;

Fig. 38 is an isometric view of the slave rotor;

Fig. 39 is an isometric view of the rear portion of the slave rotor;

Fig. 40 is a cross-sectional view of an angle back face rotor assembly taken at line 33 – 33 in Figure 42;

Fig. 41a shows a pressure distribution acting upon the rotor taken at line 41 – 41 in Figure 49;

Fig. 41b shows a resultant force acting upon the rotor taken at line 41 – 41 in Figure 49;

Fig. 42 is a schematically top view of the mean surface area acting upon a rotor in the radial plane;

Fig. 43 is a schematic sectional view illustrating the pressure distribution upon a rotor;

Fig. 44 illustrates the resultant force based upon the pressure and surface area orientation and the advantageous rotation about the centerpoints of the rotor;

Fig. 44a and Fig. 44b illustrates the benefits of having the tapered back face where an even wear is more likely to occur;

Fig. 45 is a bottom view of a rotor showing the resultant force acting upon the rotor;

Fig. 46 illustrates the resultant force acting upon the outer surface of a rotor;

Fig. 47 shows the tapered back face surface of a rotor with a plurality of conduit openings and the force distribution thereupon;

Fig. 48 is a side view showing the high-pressure side of a tapered back face illustrating a resultant force acting thereupon;

Fig. 49 is a side cross-sectional view of a slave rotor;

Fig. 50A is a front cross-sectional view of a rotor with and annular recessed portion to allow a high-pressure distribution on the low-pressure side of the rotor;

Fig. 50B is another front cross-sectional view of a rotor with and annular recessed portion to allow a high-pressure distribution on the low-pressure side of the rotor showing various dimensions;

Fig. 51 is an isometric view of an end cap that allows an in-parallel flow combination to have the rotor sections be inverted one hundred eighty degrees about the shaft;

Fig. 52 shows an in-parallel flow arrangement utilizing the cap as shown in Fig. 51;

Fig. 53 shows another variation of a cap that can be used in an in-series combination flow or alternatively and in-parallel flow pursuant to the embodiment shown in Fig. 56;

Fig. 54 shows a modified form to the cap as shown in Fig. 53 having radially extending ribs;

Fig. 55 is an outside view of the casing showing various possibilities of exit ports to allow radially exiting flow from the apparatus;

Fig. 56 is a sectional view with the cap from either Fig. 53 or Fig. 54 connecting the two rotor sections illustrating the possibility of radial ports;

Fig. 57 shows another embodiment of the present invention that is purely axial flow having axial ports that enter into the operating chambers of the rotor assembly;

Fig. 58 discloses a radial flow entrance and exit embodiment where the rotor assemblies potentially have discrete fluids passing therethrough and are both rotated by a common shaft;

Fig. 59 is a top view of another version of a section that is a portion of the casing;

Fig. 60 is a cross-sectional view taken at line 60 – 60 of Fig. 59;

Fig. 61 is an isometric view of the inward portion of the casing section;

Fig. 62 is a rearview of the casing section as shown in Fig. 61;

Fig. 63 is a cross-sectional view taken at line 63 -- 63 of Fig. 59;

Fig. 64 is front view of another variation to the rotor assembly where the lobe widths vary at different radial angles;

Fig. 65 is front close up view of the rotor assembly of Fig. 64 at bottom dead center where the lobe widths vary at different radial angles;

Fig. 65 is front close up view of the rotor assembly of Fig. 64 at bottom dead center where the lobe widths differ at alternating radial angles where the slave lobe is at bottom dead center.

Detailed Description of the Preferred Embodiment

Throughout this description reference is made to top and bottom, front and rear. The device of the present invention can, and will in practice, be in numerous positions and orientations. These orientation terms, such as top and bottom, are obviously used for aiding the description and are not meant to limit the invention to any specific orientation.

Points on a line bisecting the larger angle formed between offset intersecting axes A and B in the plane defined by axes A and B will be referred to as being at the "top", while points on the extension of that line bisecting the acute angle between axes A and B will be referred to as being at the "bottom".

To ease explanation, two axis systems are defined, one for a first rotor (power rotor) and a second axis system for a second rotor (slave rotor) where the angle between the axis is defined as α (see Fig. 11). The axis 10a is referred to as the longitudinal or axial power axis and is defined as the center of rotation of shaft 30 for the power rotor 26. Likewise, the axis 10b is referred to as the longitudinal or axial slave axis and is defined as the center of rotation for shaft 80 of the slave rotor 28. The axis indicated at 12a is referred to as the power radially lateral axis and axis 12b is referred to as the slave radially lateral axis. Further, the arrow 14a is the power radially transverse axis pointing in a forward direction and the arrow 14b is a slave radially transverse axis indicating a forward direction. The origin of the axis systems is most commonly (and by default) at the common center of the rotors.

The foundational description of the formation of the rotors is discussed below with reference to Figs. 1-8 followed by a description of the first embodiment employing two vanes to comprise a lobe with opposing contact faces. The description then

discloses an axial flow single shaft design followed by a multi-stage design. Finally there is a discussion regarding transverse balancing of the rotors with reference to Figs 37-50 and other variations of the embodiments following.

The engine (machine to convert energy, including a pump, external combustion engine, motor, turbine, compressor, flow meter etc.) 20 as shown in Fig. 1 comprises a housing 22, a bearing 23, and a rotor assembly 24.

The construction of the rotor assembly 24 is described referring to FIGS. 1A, 1B and 1C, where a rotor 26 is shown, for example, in an initial stage of construction. The slave rotor 28 of FIG. 9 is constructed in a similar manner. The master rotor 26 begins as a sphere with a shaft 30 lying along an axis A. Axis B is shown at an angle α to the axis A.

Referring to FIGS. 2A, 2B and 2C, material is removed from the master rotor 26 to leave a conical funnel section 25 whose apex is at the center E of the spherical master rotor 26, and whose apical angle is 180°-α. The axis of the funnel section 25 lies along the axis 10a.

Referring to FIGS. 3A, 3B and 3C, a conceptual cone 27 is overlaid on the master rotor 26. This conceptual cone may be thought of as part of the slave rotor 28 plus a desired gap clearance, as if the conceptual cone 27 were lying on the equivalent part of the slave rotor 28 when the slave rotor 28 has its center located at the center of the master rotor 26 (both at center E of the spherical housing). As shown in FIG. 9, the conceptual cone 27 is the tip 93 of one of the lobes 84 of the slave rotor 28 minus the desired fluid film gap radial distance. In one form, the cone 27 has its apex at the center of the sphere of the master rotor 26, and its central axis C lies along the surface of the conical face of the

master rotor 20, such that the central axis C is a radius extending outward from the center of the cavity at an angle $\alpha/2$ from a normal to the axis of the other rotor. In effect the central axis C of the cone bisects the larger of the two angles formed by the axis A and the axis B in the plane in which both axes 10a and 10b lie. As shown in Fig. 4D, the cone 27 has an apical angle ψ. The value of ψ depends partially on the strength of the material of which the master rotor 26 and slave rotor 28 are made. The greater the angle ψ, the lower the stresses on the tips of the vanes 35 and 37 that comprise the lobes 32 and 82, and the lower the pressure exerted by the lobes 32 and 82 on the engagement 36, 38, 88 and 90 (see Fig. 9). Large values of α near 45° requires smaller values of ψ to avoid the vanes extending past the axis of rotation and to avoid removal of too much material, the material being needed to support the vanes. Smaller values of α may have larger values of ψ for like reason where α is between 0+° and 45°.

Referring back to Fig. 4D, the apical angle ψ of the conceptual cutting cone 27 is equal to the apical angle $\psi 1$ of the adjacent engagement tip of the opposing rotor that engages the face to be cut by the cone 44 plus the required apical angle $\psi 2$ to create a desirable fluid film gap. The additional angle to create a fluid film gap can actually be a negative value whereby the cutting cone has a smaller apical angle then the adjacent engagement tip to create an interference fit. This is desirable when the rotors are made from materials that have a lower modulus elasticity and an interference seal is desired where the lobes of opposing rotors are forcefully engaging one another.

As shown in Fig. 9, the tip 91 of the opposing rotor that engages the contour face 36 is referred to as the adjacent engagement tip with respects to the face. An adjacent tip with a

corresponding contour face are collectively referred to as a "tip-face combination". As shown in Fig. 16, two adjacent opposing rotor contour faces such as the first surfaces 36a and 88a for the first and second rotors 26 and 28 are collectively referred to as "adjacent contour faces of opposite rotors". Further, the rearward and forward surfaces such as surface 51a of the second rotor 28 and the forward surface 42a are collectively referred to as a "slave rearward surface -- power forward surface combination". In a similar manner, the combination of opposing rotor surfaces for the slave forward surfaces and the power rearward surfaces such as 76a and 92a are collectively referred to as a "slave forward surface -- power rearward surface combination".

The value of $\psi 2$ can change with respects to ρ (the distance from the common center of the rotors) to create a variable fluid film gap with respects to ρ. For example, the angle $\psi 2$ can increase with respects to ρ to create a consistent fluid film gap from the radial outward portion of the contact face to the radially inward portion of the contact face. Likewise, the value of $\psi 1$ can change with respects to ρ because the points of engagement between points on a tip face combination the opposing rotors is a constant distance from the common center of the rotors. In other words, as shown in Figs. 3C and 4A, the circular region 29 of the cone 27 is at a distance from the common center the rotors. The center of the circular region 29 defines a teardrop shaped engagement line 19 that is a constant distance from the common centerpoint E. Therefore, at any given distance ρ_i from the centerpoint, the cutting cone can have a variable radius from the central axis C of the cone. Thus, the cone can have a variety of shapes even a straight cylinder or even an inverted cone.

Referring to Figs. 4A, 4B and 4C, to create a contour face (or surface) 38, the conceptual cone 27 is rotated with the master rotor 26 as if the cone were on the slave rotor 28 lying on axis B with its center at the center of the master rotor 26. The path of the cone 27 is shown in Fig. 4A where the travel of the center axis C is defined as the base reference line or teardrop surface 19. The locus L of the center of the cone at the surface of the rotor 26 in the frame of reference of the master rotor 26 is shown in Figs. 5A and 5B. Fig. 5A shows a top view. Fig. 5B shows a view along the axis A where it will be seen that the locus L is a tear drop shape. The actual shape 38a removed by the cone 27 is defined approximately by adding a band $\psi/2$ wide around the tear drop shape shown in FIG. 4A. The tear drop is on the surface of a sphere so that angular distances are readily calculated.

A mathematical description of the locus L is as follows.

If ρ is the radius of the sphere defining the master rotor 20, and θ is the rotational angle from the top, and α is the angle between the axes 10a and 10b then the trace of a point (x,y,z) on the axis 10b in the frame of reference of the master rotor 20 for the base reference tear drop 17 has been found to be:

$$X\_base = \rho\left(\cos(\theta)^2 \cos\left(\frac{1}{2}\alpha\right)\cos(\alpha) + \cos(\theta)\sin\left(\frac{1}{2}\alpha\right)\sin(\alpha) + \cos\left(\frac{1}{2}\alpha\right) - \cos\left(\frac{1}{2}\alpha\right)\cos(\theta)^2\right)$$

$$Y\_base = -\rho\sin(\theta)\left(\cos(\theta)\cos\left(\frac{1}{2}\alpha\right)\cos(\alpha) + \sin\left(\frac{1}{2}\alpha\right)\sin(\alpha) - \cos\left(\frac{1}{2}\alpha\right)\cos(\theta)\right)$$

$$Z\_base = \rho\left(\cos(\theta)\cos\left(\frac{1}{2}\alpha\right)\sin(\alpha) - \sin\left(\frac{1}{2}\alpha\right)\cos(\alpha)\right)$$

The lobes 32 have a radial outer surface 34, a first surface and second surface 36 and 38 (described above), a spherical inner surface 40, and a forward surface 42. The lobes 32 further have tips 44 and. These tips each have a contact surface 48 and 50. The surfaces 36, 38, 48 and 50 are described further herein.

The forward portion of surface 34 defines at least part of a sphere and is adapted to engage the inner surface of the housing 22 (see Fig. 1). The spherical inner surface 40 also defines a portion of a sphere and is intimately engaged a center bearing 23 or is a unitary piece with a center bearing 23 as shown in a second embodiment below. In the broader scope the surfaces can converge to the centerpoint of the rotors whereby removing the need of the center bearing 23.

The surfaces 36 and 38 (as well as the forward portions of surfaces 88 and 90) are described above and thoroughly in U.S. patents 6,036,463 and 5,755,196 which are hereby fully incorporated by reference.

The surfaces 36 and 38 comprise a concave and convex continuous surface with a precisely placed inflection point. As seen in Fig. 12, at the base portion of the lobe 32, the first surface 36 extends rearwardly to a first concave portion 52 then to a rearward portion 54 and the continuous surface continues forwardly to surface 56. The aforementioned concave surfaces 52, 54 and 56 are defined as the contour base surface 58.

Best seen in Fig. 12, the base surface 58 extends forwardly to an inflection point 60. At the inflection point 60, the first surface 36 transforms from a concave to a convex surface. Therefore surface 62 is a convex surface that is also adapted to receive the tip 91 of the slave lobe 82.

Consistent with the foregoing, Fig. 13 shows the second surface 38 extending rearwardly to a first concave portion 64 then

to a rearward portion 66 and the continuous surface continues forwardly to surface 68. The aforementioned concave surfaces 64, 66 and 68 are defined as the contour base surface 70. Best seen in Fig. 13, the contour base surface 70 extends forwardly to an inflection point 72. At the inflection point 72, the first surface 38 transforms from a concave to a convex surface. Therefore surface 74 is a convex surface that is also adapted to receive the tip 93 of the slave lobe 82. A more thorough discussion of the relationships between the surfaces of the master rotor and the slave rotor will follow after a thorough description of the slave rotor.

Now referring back to Fig. 9, the rearward surface 76 is positioned between contour base surfaces 58a and 70b of lobes 32a and 32b. The rearward surface 76 does not come in contact with the slave rotor 28 but cooperates with the inner surface of the housing 22 (see Fig. 1), the outer surface of the bearing 23 and the forward surface 92 of the slave rotor 28 to define an operating chamber 105 that is further discussed herein.

There will now be a detailed discussion of the slave rotor 28 where reference is made to Fig. 9. The slave rotor 28 comprises a shaft 80 and a plurality of lobes 82. As with the description of the power rotor 26, only lobe 82a (otherwise referred to as slave lobe or second lobe) will be described in detail with the understanding this specification applies to all of the lobes 82 (where 82 collectively refers to 82a, 82b and 82c). Further, the invention is not limited to the number of lobes as shown in the preferred embodiment, but the power and slave rotors 26 and 28 will have an equal number of lobes.

As seen in Fig. 9, the slave lobes 82 comprise a spherical outer surface 84, a spherical inner surface 86 (see Fig. 11), a first surface 88, a second surface 90, and finally a forward surface 92. Further, the lobes 82 has a first tip 91 and a second tip 93. The tip

91 has a contact surface 95 and tip 93 has a contact surface 97. The contact surfaces 95 and 97 are adapted to engage surface 88 of the power rotor 26.

In a first embodiment, lobes 82 are symmetrical about the radially extending plane 94 (see Fig. 9) and hence the first surface 88 will be described in detail with the understanding the geometry and other relevant features relates to the second surface 90 has a substantially mirrored image about plane 94. It should be noted that certain symmetrical variations could be employed in the lobes 82 about plane 94.

The first surface 88 is shown in Fig. 14 where the rotor assembly 22 is in a bottom dead center position. The first surface 88 comprises a concave portion 96 and a convex portion 98. The inflection point line 100 is the location where the surface 88 transforms from a concave to a convex configuration. As seen in Fig. 15, the concave surface 96 has a forward portion 99 and a rearward base portion 101. The concave surface 96 further has a loss of fluid film seal line 102. The loss of engagement line 102 defines the point where the engagement surface 50 of the vane 46 radially repositions from the surface 88 and separates (or partially separates) the tip 46 of the power rotor 26 from the rearward base portion 101. It should be noted that the contact surface 50 and first surface 88 are not necessarily in direct contact in operation, but rather there is a thin fluid film thereinbetween. Throughout this text the term engagement when directed to a contact tip and a contour face is defined as a fluid film layer between two adjacent surfaces that provides a fluid film seal and a fluid film layer where there is not a pressure difference between the engaged surfaces. Further, engagement covers an embodiment where there is an interference fit between opposing rotors where it is desired to have the rotors forcefully engage one another to create a contact type seal.

However, as a contact surface 50 rotates to the bottom dead center position as seen in Fig. 14, the annular gap 104 is produced. As shown in Fig. 15, the dashed line 106 defines a curved plane that is defined by contact surface 50 as the power rotor rotates about its axis 10a (see Fig. 11).

The perpendicular distance between curved plane 106 and rearward base surface 101 defines the annular gap 104. The distance of this gap changes with respect to the radial position θ. Hence, as shown in Fig. 16, the open area 110 is defined as the open area defined between surface 50 of the power rotor 26, surface 101 of the slave rotor 28, the housing 22 and the bearing 23. Fig. 16 shows the open area 110 has a hatched open area where the dashed line 111 indicates the perpendicular distance between contact surface 50 of vane 46 and rearward surface 101. It should be noted that rearward surface 101 has a greater radius of curvature then contact surface 50 and hence the narrowest passage between semi-chamber 113 and secondary-chamber or otherwise referred to as semi-chamber 109 (defined further herein) is open area 110. The cross-sectional area gap of open area 110 could have certain ratios with respect to the viscosity of the fluid medium that is passed therethrough. For example, if the engine 20 is designed to pump high viscosity fluids, open area 110 can be larger to allow the pressure transfer to happen quickly between semi-chambers 109 and 113. Consistent with the foregoing communication between semi chambers 113 and 121 have a similar communication means between vane 44 and the base surface 50 of the slave rotor 28.

The ratio of the distance 101 and the ratio of the contact film distance between the vane 91 and the base surface 52, 54, and 56 can be in the order of 20 to 1 in a preferred form for many fluids. This ratio is further relevant to the net cross-sectional open area

110 and the net cross-sectional area of the fluid film at location 108. A secondary range for the net cross-sectional areas can be between 30 -- 1 and 10 -- 1 and the ranges therein between and at certain ratios with certain fluids. In some cases much lower ratios can achieve the fluid pressure balancing between semi chambers. For this patent application, communication is defined as sufficient open area allowing a desirable pressure equalization between two rotationally adjacent semi chambers.

It should be noted that a very minimal amount of flow between the cross-sectional open area 110 is necessary to create a pressure balancing effect in some conditions. For example, at high speed and low-pressure, a very low ratio can create desirable balancing results.

The rotor assembly 24 comprises several chambers and semi-chambers. A chamber is defined as a substantially sealed and closed area where leakage of fluid from or to the chamber only occurs due to a passage through a thin fluid film layer between two surfaces. A secondary-chamber or semi chamber is defined as a cavity where two adjacent semi-chambers comprise a chamber; however, the open area 104 allows fluid passage thereinbetween.

As seen in Fig. 9 the forward surface 92 of lobe 82 and rearward surface 76 of the power rotor 26 along with the surface of the bearing and additionally the inner surface of the housing 22 (shown in Fig. 1) create a sealed chamber 105. Now referring to Fig. 16, the first surface 88 of lobe 82a and the first surface 36 of lobe 32 along with the outer surface of bearing 23 and inner surface of the housing 22 define the first semi-chamber 109. The forward surface 42 of lobe 32 and rearward surface 51 in-combination with the surface of bearing 23 and the inner surface of housing 22 define the second semi chamber 113. Semi chamber 121 is similar to 109

except on the opposing side of the open area between two adjacent lobes of a rotor.

As shown in Fig. 10A -- 10S there are numerous front views of the rotor assembly at single rotational position but taken at various twenty degree angle increments thereabout. Due to the rotational symmetry of each of the rotors, with a three lobe design the rotors repeat their rotational pattern every one hundred and twenty degrees (360 degrees / 3 lobes). This succession of figures is particularly useful for illustrating a fixed position for the six operating chambers 105a, 105b, 105c, 107a, 107b and 107c and further shows the formation of the sub chambers. In particular, the sub chambers 109,113, and 121 that are best shown in Fig. 16 eventually comprise the primary chamber 107during rotation from bottom dead center to top etc. The primary chamber 107c is at a near maximum volume in the top dead center proximate location in Fig. 10J. As shown in Figs. 10A and 10K it can be seen that the sub chambers are primary importance that the bottom dead center location where a fluid gap 110 is provided between the sub chambers 109a and 113a. As shown in the success of figures the chamber 107c and 107b, the sub chambers are not yet formed however, as shown in Fig. 10P the area designated at 121b' shows where a sub chamber is either beginning to form or dissolve depending upon the rotation of the rotor assembly 24.

Given the foregoing, the importance of the open surface area 110 to allow rotational balance of the slave rotor 28 will now be discussed with reference to Figs. 17 -- 18. Fig. 17 shows a front view of each of the lobes 82a -- 82c of the slave rotor 28. In this particular configuration the apparatus 20 is schematically shown in configuration advantageous for noncompressible fluid (but still very functional for a compressible fluid). The housing 22 of Fig. 1 that intimately engages the outer spherical surfaces and 34 and 84 is

shown as the hatched surface 112 and 114 at the lower and upper portions respectively. The power rotor 26 is rotating in the direction indicated by arrow 116 and hence in Fig. 17A the area to the left is a high-pressure area indicated by the letter "H" and the area to the right is the low-pressure area indicated by the letter "L". It should be noted that the high and low pressure areas are not in communication and the only means by which a fluid can pass from the low to the high pressure area is through the chambers of the rotor assembly 24.

Fig. 17 is a snapshot of the rotor configuration 22 at a specific rotational orientation with respect to the housing 22. As will be shown herein, the pressure upon first and second surfaces 88 and 90 for each of the slave lobes (i.e. 82a, 82b, 82c and 40a, 40b and 40c) have offsetting pressures and balance with respect to the central axis 10b (i.e. the slave rotor 28 has no pressure difference amongst the sum of its faces to induce a rotation). In this embodiment three slave lobes 82 are employed; however, additional lobes could be added without departing from the teachings of the present invention.

First looking at Fig. 17A, the first surface 88a is subject to a high-pressure. The high-pressure from the fluid is a result of an open circuit through open area 110. In other words, the high-pressure fluid that has access to the open semi-chamber 113 and transmits the pressure through the open area 110 to semi-chamber 109. The tip 91a is in close communication with the first surface 36a and hence provides a substantial seal between the high and low-pressure zones. The second surface 90a is exposed to the low-pressure zone.

Now referring to Fig. 17B, the first surface 88b of lobe 82b is subjected to a low-pressure zone. However, second surface 90b is subject to a high pressure zone (refer to Fig. 17C where high-

pressure fluid enters into chamber 107c at the open portion indicated by arrow 115).

Finally referring to Fig. 17C, first surface 88c and second surface 90c of lobe 82c are both subjected to high-pressure. Lobe 82c is directly in the high pressure port of the housing 22.

As seen in Figs. 18A -- 18B, there is shown a front view of the surface 112 that represents the portion of the housing that is in communication with the outer surfaces 34 and 84 of the lobes 32 and 82 respectively. The power rotor 26 is rotating in the direction indicated by arrow 116 and hence the left hand portion of Figs. 11A – 11C is designated as the high-pressure side indicated by the "H" and the right hand portion is a low-pressure side (a fluid intake side) indicated by the letter "L". The border of the high-pressure port is indicated at 117 and the border 119 of the low-pressure port each have a characteristic shape discussed further herein.

As seen in Fig. 18A, the tip 91a is in engagement with surface 58 of the power rotor 26. The tip 93c is in communication with contour base surface 70. Therefore, the semi chambers 109 and 121 have the fluid held therein under high-pressure. Therefore, surfaces 88a and 90c both have high pressure fluid acting thereon. Surfaces 90a and 88b (not shown) have low pressure fluid acting thereon. Finally, a high-pressure fluid is exposed to surfaces 88c and 90b. In summation, the opposing faces 88c and 90c are both subjected to high-pressure and hence rotationally cancel each another out. Opposing services 88a and 90b are also both subjected to high-pressure having a rotationally biasing canceling effect. Finally, opposing services 90a and 88b are both subjected to low-pressure. Therefore, out of the three leading-edge contoured faces 90a, 90b and 90c, two surfaces our subjected to high-pressure and one is subjected to low-pressure. Likewise, out of the three trailing edge contoured faces 88a, 88b and 88c, two surfaces

are subjected to high-pressure and one is subjected to low-pressure creating a rotationally balanced slave rotor 28.

Now referring to Fig. 18B the power rotor has rotated approximately 10 degrees in the direction indicated at arrow 116 and the surface 88a is now exposed to the low pressure zone "L". The fluid in semi chamber 121 is additionally in a low pressure zone because as mentioned above, the fluid is allowed to pass through open area 110a and 110b. Therefore, opposing contour faces 88a and 90c are subjected to low-pressure, and contour surfaces 90a and 88b are subjected to low-pressure as well. Finally, surfaces 88c and 90b are both subjected to high-pressure. It is important to note that leading and trailing contoured faces 90 and 88 should have an equal number of low and high pressures exposed thereon to remain rotationally balanced. As described further herein axial ports in the casing and on the rotors can be employed to provide a balancing effect.

As seen in Fig. 18C, the rotor assembly 20 has rotated several degrees further in the direction indicated by arrow 116. The vane 93c is in communication with base surface 70b. It should be noted that the tight communication with the fluid film allows very little back flow from the high-pressure zone "H" to the low-pressure zone "L".

Fig. 18D shows a rotor assembly rotated a few more degrees in the direction indicated by arrow 116 where the contact between vane 93c and base surface 70b maintains the pressure difference between the high and low-pressure zones between the opposing faces of the lobes of the rotor.

Fig. 18E now shows the slave lobe 82c positioned substantially behind surface 112 of the housing. At this position vane 93c and 91c are in tight communication with the base surfaces 70 at this position surface 90c is exposed to the low pressure zone

"L" and surface 88c is in communication with the high-pressure fluid zone "H". Further, surface 88a and 90a are in communication with the low-pressure zone "L" and surfaces 88b and 90b are in communication with the high-pressure zone "H". Therefore the tangential forces acting upon the slave rotor 26 are still balanced.

Finally as seen in Fig. 18F, the rotor assembly 20 has rotated approximately fifteen degrees in the direction indicated by arrow 116. In this position the contact seal provided by vane 93c is in the low-pressure zone "L" and the fluid film seal is between tip 91c and the base contours surface 58.

With the foregoing in mind, it can be appreciated that the open area region 104 allows communication to the lobes that are located adjacent to the casing at the bottom dead center or top dead center. Hence, the slave rotor is rotationally balanced about the longitudinal slave axis.

Figs. 19 – 21 show a second embodiment that allows axial flow of fluid. In this embodiment the aforementioned method of balancing the slave rotor could be applied as well. In general, the second embodiment allows the working fluid to substantially flow in line with the axis of rotation of the shaft 122.

Referring to Fig. 19, there is a rotor-assembly 120 comprising a shaft portion 122, a first rotor 124 and a second rotor 26. The first rotor comprises a plurality of lobes 128. Likewise the second rotor also has a plurality of lobes 130. The shaft 122 passes therethrough the first rotor 124 and does not need to be a unitary structure therewith. There are several advantages for this configuration; the first being that it expands the choice of materials for the rotor 124 in that the shaft could be a less-expensive material with different properties such as Modulus of Elasticity, hardness, rigidity, etc.

Rotation of the rotors about 180°-220° around the axes A, B, with consequential movement of the cone 27 within the master rotor 26 is required to create the entire engagement face 36. Rotation less than 180° by a small amount may be acceptable in some cases, although not preferred. Such a design may allow some fluid flow between the lobes at the bottom point of the rotation. This may avoid vibration due to rapid pressure changes in the chamber between the two contact faces at the bottom of the rotation. At this position, the contact faces lie adjacent one another. If one contact face is constructed by rotation less than 180° then the corresponding contact face on the other rotor could be constructed by rotation greater than 180°.

The cone could be rotated 360° during construction but as the surface so created prevents use of interlocking vanes, requiring subsequent removal of material from the master rotor 26, there is no need to do so. The contact faces 36, 38, 88 and 90 of each rotor 26, 28 are defined in this manner. There may be any number of contact faces on each rotor on any number of lobes that can be fit upon.

Effectively, this manner of construction means that each contact face~~ of the~~ rotor 26, 28 is defined by the locus formed as the rotors 26, 28 rotate about their respective axes A, B by points on the other rotor lying along an outer edge of the cone.

Since the contour faces 36, 38, 88 and 90 (see Fig. 9) of each rotor are defined by the movement of points on the other rotor as the two rotors rotate with each other, it can be guaranteed that there will be points of contact between the two rotors along a radially extending line R lying along a contact face through at least 180° of motion (see Figs. 7A – 7C). The lines R shading the contact face 36, 38, 88 and 90 in Figs. 4A, 4B, 4C, 6A, 6B, 6C, 7A, 7B and 7C illustrate the radial lines which define the instantaneous points of

contact as the rotors rotate relative to each other. As the line defining the points of contact between the rotors reaches its furthest penetration into the rotor, continuation of contact on that contact face will mean that the contact face will wrap back on itself as shown in Fig. 5A. This would allow no part of the slave rotor 28 to penetrate the tear drop shape, unless the opposed faces of the tear drop cavity swept out by the conceptual cone maintained a sufficient separation to allow penetration by a vane of the slave rotor. Therefore, in the case where the vanes are to be symmetrical, it is necessary for the point of contact between the rotors to switch to a corresponding contact face on the other rotor. It so happens that when each rotor is a mirror image of the other, and contact faces are defined as illustrated in FIGS. 4A, 4B and 4C, then the line of contact switches from the contact face 36 of one rotor to a contact face of the other rotor. This switch occurs at the bottom of the housing and at the top of the housing, namely when the contact faces straddle the line bisecting the acute angle between the axes A and B. As described further herein the base region of the slave rotor 28 will not engage the tip 46 of the master rotor to provide access to a secondary chamber at bottom dead center (BDC). By construction of all contact faces 36, 38, 88 and 90 in the manner described, engagement between vanes the engagement tips and the contour faces of opposed rotors may be guaranteed for at least one set of tip-face combinations. Use of a cone for shaping one rotor, thereby removing material, however, will leave a gap between the rotors unless material is added to the other rotor.

FIGS. 6A, 6B and 6C, show how gaps between the rotors at the vane contacts are avoided. A cone of material 44 corresponding exactly to the portion of the conceptual cone 27 having a apical radius of $\psi 1$(psi1) as shown in Fig. 4D is added to

the rotor. It should be noted that the values of $\psi 1$ and $\psi 2$ can be functions with respect to ρ (the radial distance from center E) and α (the amount of rotation about the axes 10a and 10b)to produce a variable tip radius from the center axis of the cone and a variable gap clearance with respects to row and the amount of rotation of the rotors. In these figures, the cone of material 27 is shown on the master rotor 26. Rotation of this cone of material in addition to the value of $\psi 2$ to define the gap clearance on the master rotor 26 while the slave rotor 28 rotates with the master rotor will create a contour face 90 (Fig. 9) on the slave rotor 28 in the same manner as the contour face 36 was created on the master rotor 26. The contour faces 88 and 90 will have the same tear drop shape as shown in FIGS. 5A and 5B. In order for the correct tear drop shape to be made, the starting point for the removal of material from the rotor must be when the axis D of the cone of material 27 lies at the top, namely along the line bisecting the obtuse angle between the axes A and B. Thus, as shown in FIG. 6A, the cone 27 must be rotated by half of its apical angle before it can be used to remove material from the slave rotor 28. The $\psi 1$ inner apical angleof the cone 27 defines the tip 44 of a vane that is part of the lobe 32 on the master rotor 26. The extra amount of material on the tip 28 created by the cone of material 48 compensates for the loss of material during construction of the master rotors contoured faces by using the conceptual cone 44. It will be noted that the cones 27 and 44 need not be exactly conical, nor must the apex of the cone be exactly at the center of the cavity, but contact portions between the vanes comprising lobes 32 of the master rotor 26 and contact faces 88 and 90 on the slave rotor 28 should have a smooth surface. The closer the apex to the center of the cavity, the smaller the clearance gap between the contact tips and engagement faces during the

operation of the rotors. The term essentially as used in the claims is intended to cover an engine whose cone 27 is not exactly defined in the manner stated, but that embodies the concept of the invention.

As shown in Fig. 6C a second conceptual cone 27a is used to define a second contact surface 36 in a similar manner as above, the center axis D' of the cone 29 travels along the teardrop shape path 31. Consistent with the foregoing, the teardrop surface 31 is formed by positioning the forward portion 33 of the teardrop at the top dead center location and simultaneously rotating the power rotor 26 about the A axis and the reference axis D about the B axis while having the reference axis D maintain a constant angle of $\alpha/2$ from the B axis. The major difference between the teardrop surface 31 and the teardrop surface 17 as shown in Fig. 4A is the direction of rotation of both the A and B axes. In a similar fashion as described above, the portion of the cone 31 having the inner apical radius $\psi 1$ is positioned in a manner as shown in Fig. 7A. It should be noted that to create the base teardrop line 31 the rotors our rotated in the opposite direction a value of about 180 -- 220 as to the direction to create the base teardrop line 17 of Fig. 4A. The second engagement surface 36 is a part of a second vane 37where the first and second vanes 35 and 37 comprise a lobe 32. In the preferred form, the material between the engagement surfaces 34 and 34 remains interposed therein between for strength and rigidity of the lobe 32.

A third engagement face 38b is created as shown in Fig. 7B. This engagement face is similar to that as shown in Fig. 6A except for the rotational location of the engagement face 38b. To put the construction of the rotor configuration in perspective, reference is made to Fig. 11 showing a completed rotor set 24 where the base

teardrop surface 17' is shown in and the center of the cone C' runs along the base teardrop surface 17' during rotation of the rotors.

The next step is removal of the material interposed between the second and third engagement faces 36a and 38b in a manner as shown in Fig. 7C. This void allows for the lobe 82a of the slave rotor to be positioned therein during operation of the engine 20 (see Fig. 9). The final step after the lobes are created is to manipulate the rearward surfaces 76 and the forward surfaces 42 to create desirable compression ratios and ensure that the forward and base surfaces of the opposing rotors do not crash into one another at BDC.

A variety of mathematical, CAD, and CNC programs can be employed to construct the aforementioned surfaces. A desirable method of making the base curve surface and the ideal curve surface can be executed to a computer-aided design program with reference to Figs. 8A – 8E. As shown in Fig. 8A, two axis are constructed and are offset from being collinear an angle α (alpha). A base reference line (bifurcating line) 39 is constructed which bisects the A 10a axis and B 10b axis. To construct the base teardrop curve for the A axis rotor, the base axis is arrayed about the A axis 10a a desirable number of increments as shown in Fig. 8B. Thereafter, the user coordinate system is set with reference to the A axis 10a and each arrayed line is rotated about the A axis 10a an equal amount of degrees as it was rotated about the B axis 10b. For example, the reference axis 39' is rotated approximately 160 degrees about the B axis. Therefore, the reference axis 39' is rotated 160 degrees back around the A axis to the position as shown in Fig. 8C. Thereafter, computer-aided drawing tools such as spline, scaling, and offset can be employed to create the base reference curve as shown in Fig. 8D. In particular, the end points of the axis as shown in Fig. 8C can be selected using a spline tool

whereby making a continuous line along these points. Thereafter, this spline can be scaled towards the origin (the common center of the rotors) and a base surface is created using the original base reference line and the scaled reference line as shown in Fig. 8E. This surface can be offset using a computer-aided drawing offset tool where the amount of offset is similar to that as the apical radius ψ of the conceptual cone 27 discussed above. Finally, a contact tip can be attached to the forward portion of the offset line 41 where the radius of the cone tip is equal to ψ1., the amount of offset between the base surface and the ideal surface minus the amount of fluid gap clearance that is desired (see Fig. 8F).

The ideal surface can be exported to a solid modeling program for constructing the forward and base surfaces and arraying and mirroring the ideal surface 41 to construct the desired number of lobes for the rotor.

With the foregoing in mind, there will now be a detailed discussion of the various aspects and components of the rotors with reference to Figs. 9 – 18 with particular attention being directed towards removal of material at the base region of the slave rotor to allow a rotationally balanced design.

As seen in Fig. 9, the rotor assembly 24 comprises a master rotor 26 and a slave rotor 28. The master rotor comprises a shaft 30 and a plurality of lobes 32a, b, and c (all of which are generally referred to herein as numeral 32). For ease of discussion, the lobe 32a will be described in detail with the understanding the specification applies to all of the lobes on the power rotor. Likewise, additional lobes could be employed without departing from the basic geometry that create sealed chambers and balanced radial forces (further discussed herein).

In this embodiment, the rotor 124 is the master rotor and rotor 126 is the slave rotor. Thereby rotor 126 would be pressed against a casing surface at indicated by the dashed line 132.

There will now be a discussion of the improvements in the conduits 131. The rotor ports 130 and 130b that are located in a high pressure portion 136 allow pressure to the rearward portion of the rotor to allow balance about the transverse axis. This balancing as described further below with reference to Figs. 36 -- 50.

Fig. 20 is an isometric front view of a rotor. The longitudinally extending ports 130 are positioned at the base portion of the lobes as well as the forward portion indicated at 130b.

There will now be a discussion of the axial flow balancing of the rotors 124 and 126. Looking at Fig. 20, there is shown an isometric view of the power rotor 124 where the dashed line 141 indicates the central axis of the power rotor (axis of rotation) and dashed line 140 indicates the axis of the slave rotor. The power rotor 124 rotates in the direction indicated at arrow 142 about axis 141. The inlet ports that are located in the casing (not shown) are indicated within the approximate range indicated at 144 and discharge or outlet port approximately indicated at range 146. To maintain balance about a vertical axis to prevent a longitudinally offset force upon the rotors (where the center of the force is either on side 144 or 146), the ports indicated by 130 and 130b allow an open circuit between the closed chambers of the engine and the chamber portion defined between the rotors and the housing. Therefore, in the right hand side of Fig. 20 (the low pressure side) the ports 130b' do not allow a pressure difference between the inner chamber and the backwall 145 and the housing. Likewise, on the high pressure side indicated at 146, there is no net pressure difference between the inner portions of the chamber and the outer

portions of the chamber. There can also be conduits through the casing to allow this pressure equalization to take place.

As seen in Fig. 20, the portions 154 and 156 represent a portion of the housing which are in communication with the perimeter surface of the power rotor 124. This ensures that the high pressure side 146 does not lead fluid back into the low pressure side 144.

As seen in Fig. 21, there is an isometric rearview of a master rotor each where the central axis of rotation is indicated at 165. In accordance with the other figures, the portion indicated at 160 is a high pressure side and the portion indicated at 162 is the low pressure side. A second embodiment of the present invention would be to remove the portion of the rotor indicated at the hash line 164. This would have the same function as the ports 130 (see Fig. 13) to equalize the pressure between the front and back portions of the power rotor 124. The seal between the high and low pressure portions is still maintained by the surfaces 167. The casing engages each of the surface 170 to maintain the pressure difference.

Referring now to Fig. 19, a plurality of rotor set combinations could be employed along the shaft 122[illegible]uld be preferable if the high and low pressure sides of each rotor set would be offset from one another by one hundred eighty degrees to prevent a moment perpendicular of the axis.

When the engine assembly 120 is used as a compressor the entrance and exit ports are located at top dead center 180 and bottom dead center 178 (or in that proximity). If the rotor set is rotating as indicated by arrow 183 and the visible side in Fig. 19 is the high pressure portion. In a compressor embodiment the housing (not shown) will cover the chambers 182, 184, 186 and 188

to allow the gas to compress therein. The compressed gas is then expelled at exit ports located in the vicinity of area 178.

In any of the mentioned embodiments in a compressor configuration the ports of the casing are positioned to the proximate location of the casing region that is shaded as shown in Figs. 11. In the compressor embodiment, a compressible fluid is contained in the chambers during the expansion and contracting phases of the rotation between the top dead center and bottom dead center areas of rotation. In a gas expander application power is extracted from the shaft or shafts based upon a pressure differential between the inlet an outlet ports. The torque from either or both shafts is employed to rotate a generator(s). The rotor assembly can be designed to be larger and hence pass a greater volume through the operating chambers whereby a lower RPM output is produced by the shaft of each rotor. Alternately, the rotor assembly can be smaller and hence a greater RPM output (e.g. 30,000 -- 120,000 and in some cases greater) can be employed with high-speed generators.

In a power generation embodiment, a generator is attached to both of the shafts 30 and 80 as shown in Fig. 9 where torque is extracted from each shaft. In this configuration rotational balance using the methods recited is not employed and two encoders tracking the exact rotational location of each shaft can be employed to control the power withdraw (thereby affecting the counter torque) to precisely position each rotor during rotation whereby the rotor tip face combinations will have minimal or no contact with one another. Without the encoders a tip face combination may be in contact; however, the force between them will be minimalized and hence minimizing wear characteristics. The compressor embodiment can be employed with a gas expander application (where the ports are preferably at bottom dead center for the inlet and starting at top

dead center for the outlet port) or alternately an incompressible fluid configuration can be employed where the sealing regions of the housing is located at top dead center and bottom dead center and a fluid (e.g. water from a dam) passes therethrough the rotor assembly. For higher pressure incompressible fluid embodiments it may still be preferable to extract torque from the power rotor and employ a rotationally balanced design described above.

One preferred method of using the rotor assembly 120 as a compressor would be to interject a fluid at the ports in the casing indicated at 180 to aid the sealing between the surfaces of the lobes 190 and the surfaces 192. The fluids primary function is to prevent leakage of gas at the contact portions at 182a and 184a and to maintain a fluid film which reduces or eliminates contact. The viscosity of the fluid inhibits the backflow of gas at these points. Further this fluid can assist in creating absolute compression.

It should be noted that the axial conduits and ports in the rotor assembly and the base housing can occur on a non central shaft design such as that shown in Figs. 1 -- 14 where the respective power shaft and slave shaft are supported by bearings at a diameter less than the diameter of the base surfaces of the rotor assemblies to provide room for the axial conduits. This design would be advantageous because the back faces of the rotors that supply a pressure force thereupon the casing and therefore do not requiring thrust bearings upon the shafts.

A fourth embodiment of the present invention is shown in Figs. 22 – 36B. The fourth embodiment assembly indicated at 220 is particularly advantageous having a modular design suited for a production model where modular sections can be placed in-parallel or in-series to produce desirable pressure and flow characteristics of the working fluid. If a higher volume of fluid is desired to pass through the assembly 220, then the modular units are placed in a

in-parallel configuration as shown in Fig. 34. If a higher pressure differential from the input portion and the output portion of the assembly 220 is desired, the modular sections are placed in a series configuration as shown in Figs. 22 where each stage increases the pressure of the working fluid with respects to the previous stage.

In general, a in-series flow configuration can be changed to a parallel flow configuration by replacing the cap 234 (Fig. 33) with cap 470 as shown in Fig. 35 to allow communication between inlet ports of 256 and 258 and outlet ports 394 and 398. In other words, the cap 470 essentially allows communication so the fluid entering can enter the operating chambers of the rotor assemblies in either the first or second rotor sections 452 and 454 (see Fig. 34).

As shown in Fig. 22, the assembly 220 comprises at least one rotor section 222, and a central shaft 224. The rotor sections comprise a casing portion 226 and a rotor assembly 228. Each rotor section 222 has an entrance portion 229 and an exit portion 231.

The casing portion 226 comprises three sections in the preferred embodiment, a first section 230, a second section 232 and a cap 234. The first and second sections 230 and 232 are commonly referred to as a base housing 231. In general, the sections 230 and 232 are adapted to engage one another at a peripheral edge and are used in the "in-series" embodiment (Figs. 22 – 23), the "in-parallel" embodiment (Fig. 34), and in the "combination" embodiment (Figs. 36A—36B).

As shown in Fig. 25, the first section (slave section) 230 is a unitary design in the preferred form and comprises a fluid entry region 240 and a fluid exit region 242, a radial inward surface 244 having a first sealing portion 246 and a second sealing portion 248 described further herein. The entrance region is defined as the

portion of the rotor assembly on the lateral side of the first and second sealing portions 246 and 248. Likewise, the exit region 242 is on the opposite lateral side of the first casing portion 230. The sealing portions 244 and 246 separate the entrance and exit regions 240 and 242 in conjunction with the outer surface of the rotor assembly 228.

Located in the upper portion of the first section 230 is the annular ridge 251 and located in the bottom portion is an annular recessed region 253.

The first section 230 further comprises a base contact surface 250 and axially extending surfaces 252. The axially extending surfaces 252 define axial conduits 254. The axial conduits 254a – 254c are located on the entry region 240 and the axial conduits 254d – 254f are located on the exit region 242.

The axial surfaces 256 located radially outwardly from the actually extending surfaces 252 define axial conduits 258. The post portions 260 defined radial conduits 262 allowing fluid to radially pass therethrough into the chambers of the rotor assembly.

The outer surface 280 is preferably cylindrical about the center point 282 as shown in Fig. 23. In a preferred form the outer surfaces sections 230, 232, and 234 comprising the casing portion 226 are in alignment in the longitudinal direction. The passageways 257 allow passage of a bolt or other connecting device (see Fig. 22).

The annular ridge 251 is adapted to be received by the annular recess region at 408 of the second section 232 described further herein.

The rotor assembly 228 comprises a power rotor 300 and a slave rotor 302 as shown in Figs. 26, 31, and 32. The rotor assembly 228 is very similar to the rotor assembly described above

in the previous embodiments where the certain elements are reiterated herein below.

As shown in Figs. 31 and 32, the power rotor 300 has an inward region 301 and an outward region 303 (see Fig. 31) and comprises a plurality of lobes 304 where each lobe has a first engagement contour surface 306, a second engagement contour surface 308, a base surface 309, and an forward surface 310. The tips 312 and 314 are adapted to engage the contact surfaces 342 and 344 of the slave rotor 302. The first and second contact surfaces 306 and 308 have an inflection point indicated at radially extending lines 320 and 322 and are constructed in the same manner as described above.

Longitudinally extending surfaces 328 define conduits 330 to allow communication between the outward region 303 and the inward (forward) region 301 of the power rotor.

Located in the central portion of the power rotor 300 is a partial sphere 332 that has an outer contact surface 333 which forms at least part of a sphere and is adapted to engage the inward surface 346 of the slave rotor 302. Located in the central portion of the partial sphere 332 is a central passageway 334 adapted to allow the shaft 224 pass therethrough and the grooves 335 adapted to engage extensions to connect to the shaft 224 in a manner so the power rotor 300 rotates with the shaft 224.

As seen in Figs. 26A and 26B, the slave rotor 302 has an inward region 339 and an outward region 341 and comprises a plurality of lobes 340 where each lobe has a first contact surface 342, a second contact surface 344, a base surface 345, and an inward surface 346. The slave rotor 302 further has a base surface 343 adapted to engage the support surface (base contact surface) 250 of the first section 230, and an inward surface 346 adapted to receive the outer surface 333 of the partial sphere 332 of rotor 300.

Each lobe has tips 348 and 350 adapted to engage the first and second contact surfaces 306 and 308 respectively in a manner to define working chambers similar to operating chambers 105 and 107 of Fig. 9. The slave rotor 302 further has an axis of rotation 360 (referred to as the offset longitudinal axis, or slave longitudinal axis) that is offset from the longitudinal axis an angle α. The surfaces 352 extend substantially longitudinally and are offset from the longitudinal axis 360 to define conduits 362 to allow communication between the inward region 339 and the outward region 341.

It should be noted in that the conduits 362 and 330 are located on both the lobes as well as the base portions. However, these conduits can be located on either portion of the power and slave rotors 300 and 302. The conduits on the lobes are referred to as lobe conduits and the conduits in the base portions are referred to as base conduits.

As previously mentioned, the rotor assembly can be similar to the rotor assemblies described above, wherein the preferred form the tips 312 and 314 of the power rotor to not engage the base surface 343 of the slave rotor to allow communication therethrough to allow a balanced rotor assembly where the slave rotor is constantly balanced about the offset longitudinal axis 360 and the power rotor 300 has a constant torque about the longitudinal axis. This is particularly advantageous for high-speed rotation rotors with high compression ratios. Alternatively, a rotor design without the balanced rotor can be employed in the axial flow embodiment particularly with low compression ratios and lower speeds.

In this embodiment the spherical portion 332 is a unitary structure with the lobes 304. Additionally, the shaft 224 can be rigidly attached to the central portion of the rotor 300. Alternatively, the spherical center portion can be a separate unitary structure

attached in to the cylindrical lobe portion of the rotor 300 by such connection methods such as where corresponding notches with a sheer member located thereinbetween holding the parts together. Any similar attachment methods can be employed with the shaft 224 and either the spherical portion 332 or the peripheral lobe portion of the rotor 300.

As seen in Figs. 27 -- 30, the second section (power section) 232 of the casing 226 is preferably a unitary member and comprises a fluid entry region 380 and a fluid exit region 382. The second section 232 further has an inward region 381 and an outward region 383. The second section 232 has a radially inward surface 384 having a first sealing portion 386 and a second sealing portion 388. The sealing portions 386 and 388 define a diameter region that separates the entrance and exit regions 380 and 382 in conjunction with the outer contact surface of the power rotor 300. The second section 232 further comprises a base contact surface 390 and axially extending surfaces 392. The axially extending surfaces defined axial conduits 394. The axial conduits 394a – 394c are located on the entry region 380 and allow communication between the outward region 383 and the inward region 381. The axial conduits 394d – 394f are located on the exit region 382 and also allow communication between the inner and outer regions 381 and 383. The radially outward axial surfaces 396 defined axial conduits 398. The post portions 400 define radial conduits 402 that allow radial communication between the radially outward region and the radially inward region of the section 232. A center cylindrical surface 404 defines a center passage 406 adapted to allow shaft 224 to pass therethrough.

The axial ports 394 have end portions 395 that can be strategically aligned at particular degrees of rotation from the top dead center location in a manner to allow passage to the operating

chambers of the rotor assembly 228 to rotationally balance a rotor therein about its axis of rotation (see Fig. 22). As shown in Fig. 18, the conduits 254 of the slave casing 230 have end portions 259 that also can communicate the operating chambers of the rotor assembly 228 to supply communication to the operating chambers to the high and low pressure regions in a strategic manner to balance one of the rotors of the rotor assembly 228. The balance is carried out pursuant to the description relating to Figs. 16 -- 18F or communication is provided to subchambers at the bottom dead center location of rotation. Therefore, this balancing action could occur on either the section 232 or 230. In the preferred form, the slave rotor 302 is balanced about its longitudinal axis of rotation and the power rotor has a constant torque applied thereupon.

The shapes of the end portions 395 of the axial conduits can have shapes such as 395' in Fig. 29 extending end portions substantially lineup to the shape of the axial ports 352 of the rotor 302 in a manner so maximum fluid flow occurs between the casing and the rotor to pressurize or depressurize the operating chambers of the rotor assembly. This allows the maximum fluid flow in a given amount of rotation of the rotor assembly.

The second section 232 further has an annular recess region 408 adapted to engage the annular extension 251 of the second section 232 and an annular extension 410 (Fig. 28) that is adapted to engage the annular recess region 426 of the cap 234. Further, a plurality of passageways 412 provide a passage of a bolt or connecting device to hold the casing portion 226 together. The passageways can be further used to allow axially extending conduits for conducting wires to pass therethrough. This is advantageous where the assembly 220 is used in a downhole pump and the driving electric motor is located below the assembly 220. Therefore, the electric wires providing electric current to pass-

through conduits similar to or exactly like 257 and 412 to allow electric current to be supplied to a driving motor (not shown).

The final component used to comprise a casing portion 226 is the cap 234. To briefly review the assembly 220, the cap 234 is used in an in-series arrangement as shown in Fig. 22 and at the end portions of an assembly 220. The cap 470 as shown in Fig. 35 is employed for the in-parallel embodiment shown in Fig. 33. The primary distinction between the caps 234 and 470, is cap 470 allows communication between the sections 222 on the input region as well as the output region.

The cap 234 as shown in Fig. 33 has a central region 420 and the peripheral region 422. Located in the peripheral region 422 is a peripheral surface 424 defining an annular slot 426 that is adapted to contact the annular extension 251 and 410 of the first and second components 230 and 232. The longitudinally extending surfaces 428 define passageways 430. The passageways 430 allow communication between the exit and entrance regions of the conduits or passageways 254, 258, 394, and 398 of components 230 and 232. A cylindrical surface 432 defines a cylindrical opening 434 is adapted to allow the shaft 224 to pass therethrough. The passageways 436 cooperate with passageways or conduits 257 and 412 of components 230 and 232 to allow bolts to pass therethrough to lock the rotor sections 222 together.

Now referring back to Fig. 22, there is shown two rotor sections 222a and 222b. To complete a functioning assembly 220, an additional cap 234c is attached to the lower portion of the assembly 220. The rotor sections 222a and 222b are substantially similar and out of phase one hundred and eighty degrees. This essentially means that two rotor sections 222 are retrieved and the outlet port 430 of the cap 234b is aligned in a manner to

communicate with the inlet ports define as both conduits 254a – 254c and 258a - 258c of the first section 230.

There will now be a discussion of the fluid flow through the assembly 220 in an in-series arrangement with reference to Fig. 24. The fluid flow is indicated by a plurality of arrows that illustrate the possible fluid paths that the operating fluid can take. It should be noted cross-sectional view shown in Fig. 22 is taken at A. rotational position as shown by line 22 – 22 in Fig. 27 where the cross-sectional view is not taken directly in line with the top dead center and bottom dead center of the rotor assemblies 228, but rather, the view is taken a few degrees counter clockwise to show the fluid flow through the radial conduits 262 and 402 of the first and second sections 230 and 232.

The pumping cycle begins with the fluid entering through the ports 430c and enters into the axial conduits 258a – 258c indicated by arrows 450a and 450b on the fluid entry region 240 of the first section 230b. The fluid indicated by arrow 450b enters into the operating chambers 109 (shown in first embodiment) of the rotor assembly 228 (see arrow 450d) or the fluid travels upwardly through the axial conduits 398 indicated at 450e and around the radially extending open regions 262 and 402 indicated at 450f and through the axial conduits 394 and through the conduits 330 of the power rotor 300 (450g). Alternatively, the fluid enters through conduits 254a – 254d indicated by arrow 450c and enters into the operating chamber 302 through.conduits 362. It should be noted that for ease explanation the rotor assembly 228 is not shown in Fig. 24. However, the operations of the rotor assembly 228 is such that the expanding operating chambers 105 and 107 (see the first embodiment) draw in the fluid and as the chambers pass the top dead center portion through the first sealing portion 246 the

operating chambers began to positively displaced the working fluid as indicated by arrows 452a, 452b, and 452c.

It should be noted that in the preferred embodiment there are three paths of travel for fluid entering and exiting the operating chambers of the rotor assembly 228; however, a functional version could use any of the conduit paths indicated by arrows 450c, 450d, or 450g for entering the operating chamber of the rotor assembly 228 and could use any combination of exit passage is indicated by arrows 452a, 452b, or 452c.

The first exit passage from the operating chamber indicated by arrow 452a passes through the conduits 362 of the slave rotor 302 (shown in figure 26N) and through the axial conduits 258d -- 258f (the axial conduits on the fluid exit region 242 of the first section 230). The fluid then passes through radial conduits 262 and 402 indicated at arrow 452d and up through axial conduits 254e-- 254g (see arrow 452e). The second exit path indicated at arrow 452b exits radially outwardly between the upper radial slots, conduit or passage 262 of the first section 230b and through the lower radial slots and 402 of the second sections 232b. The fluid 452b then joins with the fluid indicated by arrow 452e and travels upwardly through axial conduits 398d-- 398f on the fluid exit region 382 of the second section 232b.

Finally, the third path for the fluid exiting the operating chambers as the rotor assembly 228 rotates and positively displaced as the fluid contained therein has indicated by arrow 452c. For this flow schema, the fluid exits the conduits 330 of the power rotor 300 and passes through axial conduits 394 of the second section 232. Finally, all the fluid exits through port 430b.

Therefore, all of the fluid that enters through port and 430c eventually exit through port 430b. The fluid flow through rotor sections 222a and is exactly the same as the fluid flow through rotor

section 222b except the entire rotor section 222a is rotated one hundred and eighty degrees about the longitudinal axis of the shaft 224 and the fluid enters the entrance region 305 of the rotor assembly 228 and exits out the exit region indicated at 307 and exits through 430a.

It should be noted that the shaft 224 does not have to extend through the slave rotor or slave rotor casing in a single stage design or at the very end of the multistage design.

There will now be a discussion of the in-parallel version of the fourth embodiment with reference to Fig. 34. The assembly 450 comprises a first rotor section 452, a second rotor section 454, the cap 234d and the shaft 224a. The in-parallel flow assembly 450 has a low pressure region 451 and a high pressure region 453. The low and high-pressure regions 451 and 453 are separated by the vertically extending plane defined by the diameter 484 of the cap 470.

In general, the in-parallel embodiment uses the same first and second sections 230 and 232; however, a modified cap 470 as shown in Fig. 35 is interposed thereinbetween. The cap 470 essentially allows fluid passage through two sets of ports 478 and 482 to enable parallel fluid flow as described further herein.

As seen in Fig. 35, the cap 470 has many of the same components as the cap 234 having a central region 472 and a peripheral region 474. A first set of longitudinally extending surfaces 476 defined a first passageway 478. A second set of longitudinally extending surfaces 480 define a second passageway 482. The cap 470 has a diameter indicated at 484 which defines a first portion 486 on the left-hand side of Fig. 35 and a second portion 488 located on the right hand side. It is important to note that the first and second passageways 478 and 482 are located on either the first or second portions 486 or 488. Although shown in

Fig. 28 three separate ports comprising two passageways 478 and 482, the important aspect of the cap 486 is that the passageways on the first and second portions 486 and 488 do not communicate with one another in order to provide a pressure differential from the incoming and outgoing fluid described further herein.

Now referring to Fig. 34, the assembly 450 has a first rotor section 452 that is the same as the rotor section 222a as shown in Fig. 24. Further, the cap 234d is the same as cap 234 as shown in Fig. 33. However, the cap 470, also referred to as an interior cap, is positioned between the base housing 231c of the first rotor section 452 and the base housing 231d of the second rotor section 454 (also referred to as an interior rotor section 454). The rotor sections 452 and 454 are collectively referred to as a parallel assembly 455.

There will now be a discussion of the parallel fluid flow through the assembly 450. Fig. 34 discloses two rotor sections 452 and 454. However, as will be come readily apparent herein, a number of interior rotor sections 454 can be employed increasing the volumetric flow throughput of the assembly 450. Further, as described further herein, the in-combination assembly utilizes an arrangement of base housings 231, rotor assemblies 228 and caps 234 and 470 to create a combination of parallel flow and serial flow the fluid and.

As shown in Fig. 34, the fluid enters through the passageway 430d and this fluid flow is indicated by arrow 500a. A portion of this fluid indicated by arrow 500b enters through axial conduits 254d and through the conduits 362 of the slave rotor 302, and the rest of the fluid travels through the axial conduits 258d indicated by arrow 500c. A portion of this fluid 500d enters the operating chambers (see operating chamber 105 and 109 of the first embodiment) of the

rotor assembly 228 and enters the high-pressure region at 453 of the assembly 450.

The portion of the fluid that passes to the high-pressure region 453 exits the operating chamber through arrows indicated at 502a, 502b, and 502c. The exit paths are similar to the exit paths indicated by arrows 452a, 452b, and 452c. The remainder of the fluid passes through axial conduits 398 and 394 and pass through the passageways 478 of the cap 470. Thereafter, the fluid passes through the rotor assembly of rotor section 452 in a similar manner as the rotor section 454.

On the high-pressure side of the assembly 450, the fluid exiting the rotor assembly of the rotor section 452 mixes with the discharge fluid from the rotor assembly 454 and the entire fluid exits through passageway 430e of the cap 234e. Thereafter, the fluid is transported to the desired location at a higher pressure than as it entered through passageway 430d of cap 234d. It is further possible to remove the cap 470 interposed between two base housings 231 where shown in Fig. 34 the fluid exit region 382 of the master section 232d would directly communicate with the fluid entry region 240 of the slave section 230c of the adjacent casing section 231c. In a similar communication manner, the fluid exit region 382 of the master section 232d would directly communicate with the fluid exit region 242 of the adjacent slave section 230c of the adjacent casing section 231c.

With the foregoing in mind, it can be appreciated that two parallel assemblies are retrieved and stacked upon one another with the shaft 224 passing therethrough the center portion as shown in Fig. 36A. The parallel assemblies 455a and 455b are stacked upon one another in a manner so the fluid entering through passageway 430f passes through the rotor assemblies 228e and 228f to the high-pressure region 453a of the parallel assembly

455a. Thereafter, the fluid passes through passageway 430g to the low-pressure region 451b of the second parallel assembly 455b. The fluid passes through the rotor assemblies 228g and 228h to the high-pressure zone 453b and thereafter exits through passageway 430h. It should be noted that that the high-pressure and low-pressure zones 451 and 453 are located on opposite sides about the shaft of each successive stage for each parallel assembly stage.

Figure 36B shows a parallel assembly 455c and 455d where three rotor assemblies 228 are employed with two caps 470 are positioned between the three base housings 231 for each parallel assembly 455c and 455d. Of course, any number of intermediate casing portions 227 (comprising a first and second sections 230 and 232) can be employed to create a multi-rotor combination parallel and in series flow arrangement. A parallel section is defined as any integer number of rotor sections aligned in a parallel flow configuration.

It is important that there is a consistent volumetric flow for each parallel flow configuration aligned in series for each parallel flow assembly to do the approximate same amount of work. Therefore, if each rotor assembly has a similar angle α and hence having the same fluid displacement per rotation, each in-serious parallel flow configuration will have the same number of rotor section 222. However, if the angle α is increased in a rotor assembly 228 in a rotor section to increase the operating chamber size and hence increase the volumetric flow for rotation or other modifications to the surfaces to allow different volumetric throughput for rotation, a less number of rotor sections would be required in that parallel flow assembly with respects to the other parallel flow assemblies with smaller angles α.

As shown in Figs. 37 – 47, a fifth embodiment of the present invention is shown. In general, this embodiment allows a balancing about the slave radial transverse axis indicated at 530.

As shown in Fig. 40, the assembly 520 comprises a rotor assembly 522 and a casing 524. The rotor assembly comprises a power rotor and a slave rotor 528. Only the slave rotor is shown for exemplary purposes where the balance about the transverse axis also applies to the power rotor. The assembly 520 has an axis system 529 comprising a slave radially transverse axis 530, a slave lateral axis 532 and a slave vertical axis 534. The axis system 529 intersects at a centerpoint indicated at 530 which coincides to the center of rotation of the slave and power rotors. As shown in Fig. 40, there is shown in a cross-sectional view where a portion of a slave rotor 528 where on the left-hand portion of the vertical axis 534 is a high-pressure zone indicated at "H" and on the right hand portion is a low pressure zone indicated by "L".

The casing 524 comprises a first section 524a and a second section (not shown). The first and second sections are very similar to the section 230 and 232, except the base surface 545 has a different radially outward slope to support the surface 544 of the slave rotor (see Fig. 37). The first section 524 comprises an annular base 529 and a plurality of radially extending connectors. The annular base 529 has a base surface 545 adapted to engage the base surface 544 of the slave rotor 528 and as shown in Fig. 40, the base surface 545 comprises a radially inward portion 525a, a radially outward portion 525b, and an outward surface 525c. The annular base 529 further has an upper surface 533 and a radially outward surface 535. The longitudinally extending surfaces 537 on the radial connectors define radially inward passageways 539. Likewise, the upper surface 533 and the radially outward surface 535 comprise a passage 541.

Fig. 49 shows a side view of the slave rotor taken in the slave radially lateral direction, where the radially transverse portion referred to as the top dead center (TDC) portion 548 is located on the left-hand portion of that figure. On the diametrically opposed region, the bottom dead center (BDC) portion the 550 is located. As referred to above, the bottom dead center portion 550 is the region where the operating chamber is enclosed and at a minimum volume. Likewise, the top dead center portion 548 indicates a location where a operating chamber is at a maximum volume.

The slave rotor 528 and the power rotor 526 are substantially similar and hence the base surface 544 of the slave rotor will be described in detail with the understanding the specification is relevant and applies to the power rotor as well.

The slave rotor 528 comprises a plurality of lobes 542 that have the properties which are is very similar to the lobes discussed above. However, the base surface 544 is angled with respects to the radial axis. Further, the outward surface 531c is angled with respects to the slave longitudinal axis 534. For purposes of explanation, the base surface on the high-pressure side is referred to as 544H and the base surface on the low-pressure side is referred to as 544L.

As shown in Fig. 39, base surface 544 has a radially inward portion 545 and a radially outward portion 547. The longitudinally extending conduits 549 allow passageway between the inward region 339 and the outward region 541 and extending through the lobe portions and the base regions respectively.

In order to best understanding the balancing of the slave rotor 528, initial reference is made to Fig. 42 which is a top view schematically shown in the cross-sectional of the slave rotor in the slave longitudinal direction. The area indicated by 560 which is located in the slave radial plane, indicates the average high-

pressure surface area acting upon the slave rotor in the radial plane. Likewise, the area 562 represents the radial plane of the low pressure region of the slave rotor. In operation, as the rotor assemblies rotate the high-pressure zones would shift about the axis 530a and numerals 530b depending upon the position of the chambers with respects to the sealing surface (see Figs. 17A – 18E). As shown in Fig. 43, the average force acting in the slave longitudinal direction as a result of the pressure in the high-pressure region multiplied by the surface area 560 is indicated by force vector 564. Likewise, the average forces acting in the slave longitudinal direction that is a product of the pressure in the low-pressure zone multiplied by the surface area 562 results in a force vector 566.

As shown in Figs. 41 and 43, the mean pressure acting upon surface 544b is indicated at 566. The mean force vector 588 is the sum of the area 566 multiplied by the pressure to get the mean force acting upon the line 589 (see Fig. 47). The force vector 588 actually acts upon the annular line 590 as shown in Fig. 45. Therefore, the force vectors 588 (an infinite number of two-dimensional force vectors acting upon annular line 590) are summed and indicated by force vector 592 the force vector 592 is a perpendicular distance from the transverse axis 530 a distance indicated at 594 (see Fig. 41).

The pressure acting upon outward surface 546 is indicated by pressure distribution 594. The sum of this pressure multiplied by the surface area is indicated by force vector 596 acting upon annular line 598. In a similar analysis as force vector 588, shown in Fig. 46, the force vector 596 is an infinite number of vectors acting upon the annular line 598. The sum of the factors acting upon the line is a resultant vector 600.

With the foregoing vectors in mind, namely 564, 592, and 600, a moment analysis about the transverse axis 530 can be conducted. It is a well-known in engineering disciplines that a moment is a force times a perpendicular distance about a point or axis. For our analysis we will be concerned about the forces acting in the plane defined by the slave radially lateral axis and the slave longitudinal axis about the slave radially transverse axis 530 (which extends straight out of from the page in Fig. 40B). The force vector 592 which only has axial components in the slave radially lateral direction and the slave longitudinal direction is a distance 594 from the slave radially transverse axis 530. Therefore distance 594 multiplied by force vector 592 creates a clockwise moment about transverse axis 530 in Fig. 41B. The resultant force vector 564 acts substantially downward in the slave longitudinal axis and is a perpendicular distance 565 from the transverse axis 530 and the product of distance 565 and force vector 564 results in a moment in the counter clockwise direction (see Fig. 43).

To understand the balanced improvements of the conical surface 544, reference is now made to Fig. 44 where there is shown a rotor 610 having a conceptual base surface 614 and a conceptual base surface 614 is similar to the base surface 544 of the rotor 528. If the rotor 610 adopted the surface 612, the resultant force based upon the pressure in surface area would be aligned somewhat close to force vector 616. By extending the line of force of force vector 616 as shown, the perpendicular distance about the slave radially transverse axis 530 is indicated at 618. However, by having a base surface 614, a resultant force 620 is produced and the perpendicular distance from the line of force of vector 620 is indicated at 622. It is graphically shown in Fig. 44 that distance 622 is greater in length then distance 618. Therefore, the surface 614 will inherently create a greater moment about the slave radially

transverse axis 530. Further, there shall conduits in the slave rotor allow for fluid to be partially dispersed upon the conical back face 525 of the casing portion 524. A similar analysis can be conducted with resultant force vector 600. The angle of the conical backface 544 where it slants rearwardly with respects to the radial plane of the rotor with respects to traveling radially outwardly is referred to as a positive angle or positive conical angle. Further, the corresponding angle of the back face 525 is referred to as a positive angle or positive conical angle.

It should be reiterated that the base surface 544 analysis is relevant to the power rotor 542 about the reference axis for the power rotor (e.g. the longitudinal power axis, the power radially lateral axis, and the power radially transverse axis). It is very desirable to have the counteracting moment resulted from the pressure acting upon the base surface 544 to prevent unnecessary wear thereon.

The second benefit of having the base surface 544 a tapered back face where it is angled with respects to the radial location, when the force vector 564 applies a moment about the slave radially transverse axis 530, the base surface 525 is better adapted to handle this rotation than that surface 525 as shown in Fig. 44A. As seen in Figs. 44, 44A, and 44B, the extension 626 of surface 612 is closer to the slave radially transverse axis 530 than the extension 626 of the mean plane of surface 612. Therefore, the surface 614 is better adapted to evenly distributed the pressure along surface 614. As shown in Fig. 44A it can be shown that the rotating action 630a of the rotor 528 about the transverse axis 530 will cause the radially outward portion 547 to hit the radially outward portion of the casing first causing additional wear at this radially outward location for the rotor and the housing. Now referring to Fig. 44B it can be appreciated that as the rotor 528 rotates a slight

amount in a manner indicated by arrow 630b about the transverse axis 530 the surface 614 is better adapted to evenly distribute pressure thereupon upon the base surface 525. In other words, the distance separating between the surfaces 614 came and 525 are substantially uniform (at least much more than in Fig. 44A) with respects to the distance radially outward from the transverse axis 530.

A further advantage of having a tapered back face as shown in Fig. 44B is that a greater surface area extending from the central shaft is created allowing a greater resultant force acting about the transverse axis 530.

It is desirable to have a mean surface angle for surface 544 with respects to the plane defined by the longitudinal axis (the plane in the radially lateral and radially transverse axis) an angle between 10 – 50 degrees. A more desirable angle would be in the range between 20 -- 40 degrees. The preferred angle is in the proximity of 30 degrees with respects to the plane defined by the respective longitudinal axis (for the power or slave rotor respectively).

Fig. 47 shows a more inclusive method of calculating resultant force vector 592. As mentioned previously, the force vector is 588 are a summation of the pressure force along the surface 544 and radial lines 589. For example, the force vector 588a is of lower magnitude then the adjacent force factors. This is because the pressure line 589a is of smaller magnitude because the passes therethrough the conduits 549 and 551 of the surface 544. The resultant force vector 592 extends in the longitudinal slave axis and slave radially lateral axis directions at the approximated angle shown in Fig. 48 of course the pressure acts upon the axial conduits of the rotors.

The force vectors disclosed in the preferred embodiment are for exemplary purposes illustrating the fundamental concepts of

having a desirable tapered conical back face. The force vectors are for explanation purposes so the reader may better understand the fundamental concepts. The force vectors are no way intended to limit the invention whatsoever, but rather are intended for an analysis to appreciate the moment that is created about the transverse axis 530. It should be reiterated that the exact position and magnitude of the force vectors will alter with respects to certain degrees of rotation of the rotors and various pressure differentials between the high-pressure port and the low-pressure port; however, the figures disclosed are intended to illustrate the general aspects of having the conical backface 544.

Fig. 50A shows an adaptation to the fifth embodiment where an annular notch portion 604 is removed from the rotor 528. This embodiment allows fluid to annularly pass around the rotor 528 to create a high-pressure resultant force 610. As shown on the high-pressure side "H" the conduit 604 will pass a portion of the high-pressure distribution 606 annularly around to create a high-pressure distribution 608 on the low-pressure "L" portion. A resultant vector 610 is a product of the surface area of the rotor multiplied by the mean pressure distribution 608. This resultant vector 610 is desirable for certain pressure scenarios where a counter torque about the radially transverse axis 530 is desired.

Fig. 50B shows a modification to the second embodiment, the various dimensions can be calculated with respects to the radius 'r' of the rotor. In a preferred form the angle A_o is equal to or greater than α so there is not an under cut in the slave casing for simpler manufacturing. The length l_{b2} is a function of the r, the angle A_o, α and the diameter d of the throughput shaft. For example, with a smaller value of d and α the value of l_{b2} can be very large which can be beneficial to supporting the rotor 528'.

Therefore the overall length can be that of ten times the value of r in more extreme circumstances or a fraction of r where the tips of the opposing rotor define the value of l_{b2}. The annular conduit width lc can have a large variety of lengths and thickness to achieve the conduit path to from the high to low pressure ports. The annular path can be on the rotor or on the casing. A variety of conduit paths can be employed, including an external hose with a dynamic restricting system that varies the flow resistance to provide a desirable pressure balance. The angle Ai of the mean surface 632 has a resultant force with a longitudinal component and a radial component that provide counter balancing forces. Of course the back face can have a variety of shapes and continuous curved contours where the effect of the backface is a balanced rotor about the transverse axis.

It should be noted that the slave rotor is not supported by exterior bearings it is supported by the ball on the power rotor, or it could have a ball that is supported by the concave spherical inner surface of the power rotor. Therefore, the ball, the power rotor, and the housing support the slave rotor in various combinations. Additionally, there could be support bearings upon a shaft of a slave rotor that supply partial anti-rotational support about one of the radial axis. Alternatively, a thrust bearing about the base surface could be employed.

The various components discussed above could have a Teflon coating or any conventional coating to reduce friction or has desirable wear characteristics where the contact portions of the various components slide upon one another are subject to a coating procedure. The various components can be produced by a CNC machine or cast from a mold.

As shown in Fig. 51, the cap 640 discloses an alternative method of directing the flow between rotor sections 222. In general,

this cap allows an in-parallel flow between rotor sections with the advantage of offsetting the radially lateral force upon the shaft 224 (discussed further herein below). The cap has a first longitudinal region 641 and a second longitudinal region 643. The cap 640 is similar to be previous caps 232 and 470 with the exception the cap 640 has a first passage system 642 and a second passage system 644. The cap has a transverse axis 646 that separates the high pressure side and the low-pressure side. The front surface 648 has contact surfaces 650 and 652 adapted to engage the upper and lower surfaces of the sections 230 and 232 to provide a seal from the high pressure side to the low-pressure side. Corresponding surfaces are located on the opposite longitudinal side of the cap 640.

The first passage system 642 comprises a first opening 654, a passage 655, and a second opening 656. The first opening 654 is in communication with the first longitudinal region 641. The first opening 654 has an entrance passage 658 in communication with the passage 655. The passage 655 extends to an exit passage 660 that is in communication with the second opening 656. The second opening 656 is in communication with the second longitudinal region 643 (the other side of the cap) as indicated by the dashed lines in Fig. 51.

In a similar configuration as to the first passage system 642, the second passage system 644 comprises a third opening 662 having a second entrance passage 664 that is in communication with a passageway 666. Fluid is adapted to extend to the passageway 666 through the second exit passage 668 and exit through the fourth opening 670. The fourth opening 670 is in communication with the second longitudinal region 643 and the third opening 662 is in communication with the first longitudinal region 641.

The cap 640 is symmetrical about the radially transverse extending line 646 and a manner so if the cap 640 was rotated one hundred eighty degrees about axis 646 it will, in the preferred form, look exactly the same as shown in Fig. 44. Any number of implementations for passageways 655 and 666 can be employed, the important aspect of the passageways 655 and 666 is a pressure differential is maintained between the high pressure side and the low-pressure side.

As seen in Fig. 52, the assembly 220a the rotors configured a similar manner as the in-series flow as shown in Fig. 24, however the cap 640 is used to join the rotor sections 222d and 222e so the flow is actually an in-parallel flow arrangement. The fluid and enters into the entrance port 672 and can in general take two separate paths.. First, the fluid can enter into the operating chambers of the rotor assembly 228e as indicated by arrows 674a, 674b, and 674c. Alternatively, the fluid can pass through the first opening 654 of the cap 640 as indicated by arrow 676a. Thereafter, the fluid would pass through the passageway 655 and exit through the second opening 656. Thereafter, the fluid enters into the operating chambers of a rotor assembly that are not shown but located at location 228d indicated by arrows 678a, 678b, and 678c. Finally, the fluid exits from the rotor assembly 228d as indicated by arrow 680 and exits the exit port 682.

The fluid exiting the rotor assembly 228e indicated by arrows 684a, 684b, and 684c enters into the third opening 662 as indicated by the dashed arrow 686a and passes through the passageway 666 (see Fig. 51) and exits through the fourth opening 670 as indicated by the dashed arrow 686b as shown in Fig. 52. Thereafter, this fluid is substantially similar in pressure to the fluid exiting the operating chambers of the rotor assembly 228d and mixes with this exiting fluid to exit through the exit port 682.

There will now be an analysis of the moments substantially about the radially transverse axis of the shaft 224. It should be reiterated that the Fig. 52 is not taken exactly along the radially transverse axis because that view would not illustrate the radial flow into and out of the operating chambers as indicated by arrows 678b and 674b, and 684b and 680b.

As discussed above, in a pump configuration, the torque about the shaft 224 creates a high pressure side in the rotor section 222e indicated by H' and a high pressure side in the rotor section 222d indicated by H" (see center portion of Fig. 52). Therefore, a resultant force 688a and 688b results on the shaft. In many implementations the offsetting forces 688 or desirable than having both of these forces aligned and distributing a lateral force on the shaft in the same direction. Therefore, an additional assembly similar to that of 220a has shown in Fig. 29a can be attached in an upper or lower portion along the shaft 224 to create an in-combination flow and each section 222 will have resultant forces 688 that offset one another and hence creates balance upon the shaft 224 in the lateral direction. Further, by having four sections 222 opposing one, another any moment about the lateral direction of the shaft 222 is reduced.

Fig. 53 shows a cap 700 that is adapted to direct the flow from one lateral portion on a first axial side in to the opposite lateral portion on the opposite axial side. The cap 700 has a transverse axis 702 that separates the high pressure side from the low-pressure side of the cap 700. The cap further has a first axial or longitudinal portion 701 and a second axial or longitudinal portion 703. The cap further has a first surface 704 and a rearward surface 706 the first and second surfaces 704 and 706 have transverse portions 708 and 710 that are adapted to engage the first and second sections 230 and 232 to provide a seal between the high

pressure side and the low-pressure sides. The cap 700 comprises a first opening 712 and a second opening 714. The first opening 712 is in communication with the first axial portion 701 and communicates to the second opening 714 and through passages 716 and 718. In the preferred form, the cap 700 is symmetrical about the transverse axis 702 whereby rotating the cap about the said axis one hundred and eighty degrees would still appear exactly like the cap as shown in Fig. 53.

As shown in Fig. 54, a second version of the cap 700 is employed referred to as numeral 720. This version is substantially similar as the cap 700, however, radially extending ribs 722 are employed having tangential passageways 724. The openings 712a, and 712b, and 712c or similar to the openings 714a, 714b, and 714c.

The cap 700 can be used for an in serious flow arrangement where the lateral pressure upon the shaft would be more similar to an in-parallel flow using the modified cap 470. Alternatively, the cap 700 or 720 can be used in an in-parallel flow arrangement as shown in Fig. 56.

As seen in Fig. 55, the assembly 220c has a plurality of exit ports 730, 732, and 734. Now referring to Fig. 56, the fluid enters into the entrance region 738 as indicated by arrow 736. Thereafter, the fluid enters into the operating chamber of the rotor assembly (not shown but indicated by numeral 228g) of the rotor section 222g by any one of the arrows 740. Thereafter, the fluid exits from the operating chambers on the high pressure side indicated by arrows 742 and exit through any one of the radial passageways 730, 732, or 734. Three radial passageways are shown for illustrative purposes where the passageway 730 remain solely on a section (the second section 232g in this example). Alternatively, the exit passage can coexist in a combination form between two sections as

indicated by passage 732. Alternatively, the radial passage can exist in combination of recess portions between a section and a cap as indicated by radial passage 734.

The other approximate half of the fluid flow will pass through the opening 712 as indicated by arrow 744a and pass through the passageways 716 and 718 (see Fig. 53) and exit through the second opening 714 as indicated by arrow 744b. Thereafter, the fluid passes to the section 222f in a similar manner as described above.

It should be noted that radial ports can be employed in a similar manner for exit and entrance regions for any rotor section alone or in-combination with the entrance and exit ports of the caps 234, 470, 640, 700, and 720.

The inlet ports can be located in the radially outward portion of either the first section or the second section or a combination thereof. The benefit of having additional inlet and outlet ports is there is potentially less fluid resistance by having the additional paths of travel.

This is very beneficial in situations where the fluid is not available in an axial flow situation, but rather only available along a radial side portion of the casing portion 226. A combination of axial and radial flow inlet and outlet ports (vice versa) can be advantageous to give increase design flexibility in many situations and applications. It should be noted that the exit ports 730, 732, and 734 are at the substantially same pressure as the exit ports 731 and all of these high-pressure ports should be confined from the entering fluid indicated at 736.

It should further be noted that in all of the above embodiments, the cap diameter of the assembly 220 can be reduced if the radially outward ports for entering the operating chambers are removed.

As shown Fig. 57, the assembly 760 comprises a first rotor section 762a and a second rotor section 762b. Rotor section 762 comprises a cap 764 a first section 766 and a second section 768 that are very similar to the sections and caps of the previous embodiments; however, the section 766 has only an axial conduit passage 770 that enters into the operating chambers through the conduits 362 of the slave rotor (not shown in Fig. 57 that as in the previous figures adapted to be mounted therein between the housing sections). The fluid exits through the conduits 330 of the power rotor on the high pressure side and passes through the passageway 772 of the second section 768. Thereafter, the fluid passes to the rotor section 762b. As with the previous embodiments a fluid with a pressure differential can pass through the entrance and exit ports of the assembly 716 and rotational work can be extracted from the shaft 224 in a motor embodiment. Essentially, the embodiment as shown in Fig. 57 removes the radial entrance and exit portions for entering and exiting the operating chambers. Of course a number of in serious or in-parallel or in-combination arrangements of the rotor sections 762 can be employed.

It should be noted that another key advantage of the embodiments disclosed in Figs. 23 and on is that a straight shaft 224 can be employed. As seen in Fig. 58 there is shown a radial flow embodiment with two rotor sections employed. Further, the caps 830 and 836 have recess portions to induce the flow in and out of the rotors.

The assembly 800 comprises rotor sections 802 where each rotor section comprises a casing 804 and a rotor assembly 806 (not shown) that is similar to the rotor assembly as described above. The casing 804 is similar to the casing described above with the exception there are radial entrance and exit ports located on the

radially lateral portions of the casing a first surface 808 defines a first radial passageway 810. The passageway (as with many of the passageways) can have a threaded recess region adapted to screw into a fluid line. Further, the exit passageway 814 is defined by a surface 812 and is located on the high-pressure region. Therefore, fluid that radially enters as indicated by arrow 816 can then into the operating chambers by either 818a, 818b, and 818c. It should be noted that the casing 804 and rotor assembly 806 can be constructed for any combination of entrance into the operating chambers using any combination of the paths 818 (of course this applies as well for the exit passage and the other embodiments). Thereafter, the fluid passes through the top dead center portion of the assembly 800 and exits on the high pressure side. The fluid can exit the operating chambers by either passage 820a, 820b, or 820c. Then, the fluid exits through the exit passageway 814 as indicated by arrow 822. A similar analysis can be conducted for fluid enters into the rotor section 802b indicated by arrow 824 and exits out of the rotor section 802b indicated at 826.

Therefore, it can be appreciated that the shaft 224 in Fig. 58 is operating two separate rotor assemblies. The fluid lines entering the ports 808 and 808a can be separate lines that are not in communication with the one another. This is advantageous if it is desired to not have the fluids mixed together but a common shaft to increase the pressure of these respective fluids (or to extract torque off of the shaft from a pressure differential from two or more fluid lines). Likewise, the exit ports 814 and 814a need not communicate with one another and can pass to separate lines or be mixed together if desired. Fig. 58 illustrates the clear benefits of having a single shaft pass through the various rotor assemblies.

A further modification in the assembly shown in Fig. 58 is the end caps have recess regions to improve the flow of fluid in and out

of the operating chambers. The top and bottom end caps 830 have semi annular recess regions 832 and 834. These regions to not extend beyond the transverse axis in order to create a seal between the high-pressure side and the low-pressure side. Alternatively, the internal sections of the casing 802 extend longitudinally into the recess region of the cap to prevent leakage from the high-pressure side to the low-pressure side. The cap 836 has opposing semi annular recessed regions 838 and 840 on opposing longitudinal sides of the cap 836. This allows greater fluid flow for both rotor sections 802a and 802b.

There will now be a discussion of a high flow section apparatus with reference to Figs. 57 – 63. In general, the section 900 has a substantial open region to allow greater fluid flow to enter the chambers of the rotor assembly (see Fig. 19). The substantial difference of the section 900 is there is a single axial surface 902 and 904 defining axial conduits 906 and 908. The large radial cross-sectional area for the conduit 906 allows low resistance fluid flow to pass therethrough. The section 900 further comprises the conduits 910 that are adapted to allow bolts or connecting devices to pass therethrough. Further, the conduits 912 can be used to pass electrical wires or other material therethrough. As with the previous embodiments the surfaces 914 define axially extending conduits 916. The radially extending ribs 918 provided additional support for the radially outward portion of the section 900. A housing component can be employed for the power rotor.

Fig. 62 shows a rearview of the section 900 where the base surface 919 is adapted to be supported upon a cap. Fig. 63 is a cross-sectional view where the surfaces 904 extend radially outwardly in the longitudinal central portion to provide a more desirable fluid flow therethrough.

As shown in Fig. 63, there is another embodiment of the rotor design. In general, the embodiment as shown in Figs. 63 -- 65 has a different width between the base surface of opposing faces of a lobe for each rotor. In other words, the rotational width of the lobes is tangentially wider for the slave rotor and the power rotor at every other lobe. The varying width of the rotors is advantageous for constructing a casing which allows the chambers and sub chambers of the rotors to come in contact with high and low pressure ports at the same radial increment of rotation whether the lobe is located at bottom dead center or the base region is located at bottom dead center.

As shown in Fig. 65, the rotor assembly 920 comprises a power rotor 922 and the slave rotor 924. The power and slave rotors 922 and 924 are constructed in the same manner as described above. The slave rotor 924 comprises a plurality of lobes 926 that have a first engagement face 928 and a second engagement face 930. A base region 929 is defined as the area between two adjacent lobes of a rotor. The first engagement face 928 is constructed from a cone where the center axis travels about the base surface tear drop 932. At the upper portion of the engagement face 928 is a contact tip 934.

In a similar manner, the engagement face 930 is offset from the base tear drop 936. A contact tip 938 is located at the upper portion of the engagement face 930.

Now referring to the power rotor 922, the power rotor comprises a plurality of lobes 940 that have tangentially opposite facing engagement face is 942 and 944. The engagement faces 942 and 944 are formed by an offset from the base tear drops 946 and 948 respectively. The cutting cone to define the engagement surface 942 is located at the center axis 939a to the contact tip 938a and has a cone radius equal to that of the contact tip 938a

plus the fluid gap offset. It can be appreciated that the center axis of the contact tips of the rotors pass through the tear drop base reference surfaces of the opposing rotor.

Consistent with the foregoing, there will now be a discussion of the rotational positions of the rotor assembly 920 at bottom dead center (BDC) and top dead center (TDC). As shown in Fig. 64, there is a front view of the rotor assembly at bottom dead center where the lobe of the power rotor 922 is in the "lobe BDC position". At this rotational position, the engagement seal points of the tip 934 and the tip 938 is indicated at engagement point 950 and 952. The tangential with between this contact point is indicated by annular distance 954. It should be noted that this annular distance 954 can also be expressed by a BDC contact point angle based upon the centerpoint of the rotors.

Now referring to Fig. 65, the base region 929 is at the BDC position referred to as the "base BDC position". The engagement point 960 and the engagement point 962 is a tangential width of 964 which can be expressed as a and angular distance from the common center of the rotors. The contact widths 954 and 964 are the same or approximately the same to allow for consistent exposure of chambers and sub chambers from the high-pressure side of the ports or the low-pressure side of the ports. In other words, the casing has ports with boundary lines that are located at rotational locations at the proximate area of the engagement points 952 and 960 for the port on the right hand side as shown in Figs. 64 and 65. As for the ports located on the left hand side with reference to Figs. 64 and 65 the boundary line of the port at bottom dead center is at the proximate location of engagement points 950 and 962.

It should be noted that many of the various corner portions through out the various embodiments have tapered corners to allow more desirable fluid flow therearound.

While the invention is susceptible of various modifications and alternative forms, specific embodiments thereof have been shown by way of example in the drawings and will herein be described in detail. It s should be understood, however, that it is not intended to limit the invention to the particular forms disclosed, but, on the contrary, the intention is to cover all modifications, equivalents and alternatives falling within the spirit and scope of the invention as expressed in the appended claims.

Therefore I claim:

1. A device to displace fluid comprising:

a housing having an inner surface;

an inner component have an outer surface that defines at least part of a sphere;

a first rotor mounted for rotation in the housing about a first axis and having a forward region and a rearward region and a first outer surface defining at least part of a sphere and adapted to intimately engages the inner surface of the housing;

a second rotor having a forward portion and a rearward portion, mounted for rotation in the housing about a second axis offset from being collinear with the first axis by an angle α and intersecting at the common centers of the rotors, the second rotor including a second inner surface defining at least part of a sphere having a common center with the center of the first rotor and is adapted to engage the said inner component, a second outer surface defining at least part of a sphere and having a common center with the second inner surface and adapted to engage the inner surface of the housing;

the said first rotor further having a first contour surface that is defined by a locus formed by points on the second rotor as the second rotor rotates about the second axis, and the first rotor further has a first engagement tip which is positioned in the forward region of the first rotor;

the second rotor further having a second engagement surface that is defined by a locus formed by points on the

first rotor as the first rotor rotates about the first axis, the second engagement surface having a base region;

the points of each rotor that define the locus lie along an outer edge of a central axis is essentially a radius extending outward from the common centers of the rotor at an angle $\alpha/2$ from a normal to the axis of the other rotor;

whereas the first engagement tip of the first rotor does not come into contact with the base region of the second rotor allowing fluid to communicate thereinbetween.

2. The device as recited in claim 1 where the second rotor is substantially rotationally balanced about the second axis with respects to the rotational position of the second rotor.

3. The device as recited in claim 1 where the first rotor has a center surface defining the said inner component and is adapted to engage the second inner surface of the second rotor.

4. The device as recited in claim 3 where the center surface of the first rotor is defines at least part of a sphere.

5. The device as recited in claim 1 where the first and second rotors have inward surfaces adapted to allow a shaft to pass therethrough.

6. The device as recited in claim 5 where a shaft extends through the inward surface of the first rotor and is attached thereto.

7. The device as recited in claim 6 where a torque is applied to the said shaft and fluid is compressed in operating chambers defined in part by the contour faces of the first and second rotors.

8. The device as recited in claim 5 where the inner surface of the first rotor engages the shaft in a manner to rotate in conjunction therewith.

9. The device as recited in claim 8 where the second rotor is substantially balanced about the second axis with respects to the rotational position of the second rotor.

10. The device as recited in claim 1 where the device is used as a flowmeter.

11. The device as recited in claim 6 where the housing has an inlet port adapted to allow fluid to enter operating chambers defined by the first and second rotor and an exit port adapted to eject fluid from operating chambers defined in part by the first and second rotors.

12. The device as recited in claim 11 where the inlet port and outlet ports are positioned on respective opposite sides of a transverse axis of the rotors.

13. The device as recited in claim 12 where the casing engages the radially outer surface of the rotors at bottom dead center and top dead center.

14. The device as recited in claim 12 where the operating chambers while located at the high pressure port are adapted to increase the pressure of a fluid contained therein.

15. The device as recited in claim 14 where the fluid is incompressible.

16. The device as recited in claim 14 where the fluid is a compressible gas.

17. The device as recited in claim 15 where the device is used as a compressor and torque is applied to the said shaft to convert energy to the said fluid in the said operating chambers.

18. The device as recited in claim 15 where the device is used as a extract energy from the said fluid where torque is used from the said shaft to convert energy from a pressure differential form the high and low pressure ports.

19. The device as recited in claim 18 where the device is a generator.

20. The device as recited in claim 18 where the device is a motor.

21. The device as recited in claim 16 where the device is used as a extract energy from the said fluid where torque is used from the said shaft to convert energy from a pressure differential form the high and low pressure ports.

22. The device as recited in claim 21 where the device is a generator.

23. The device as recited in claim 21 where the device is a motor.

24. The device as recited in claim 16 where the device is used as a pump and torque is applied to the said shaft to convert energy to the said fluid in the said operating chambers.

25. The device as recited in claim 1 further comprising:

the first rotor further having a second contour surface that is defined by a locus formed by points on the second rotor about a central axis that is $\alpha/2$ from the normal of the second axis where the first and second contour surfaces of the first rotor are on opposite sides of a lobe of the first rotor.

26. The device as recited in claim 25 where the second contour surface of the first rotor has a second engagement tip of the first rotor positioned at the forward region of the contour surface and the second engagement tip of the first rotor is adapted to engage a contour surface of the second rotor.

27. The device as recited in claim 25 further comprising;

    the second rotor further having a second contour surface that is defined by a locus formed by points on the first rotor about a central axis $\alpha/2$ from the normal of the first axis where the first and second contour surfaces of the second rotor are on opposite sides of a lobe of the second rotor.

28. The device as recited in claim 27 where a plurality of lobes are positioned at radially even increments on the first rotor and the lobes of the first rotor are adapted to be interposed between a plurality of lobes of the second rotor.

29. The device as recited in claim 28 where the second rotor has a base surface with a positive angle that provides rotational balance about a transverse axis.

30. The device as recited in claim 37 where a conduit between the high pressure portion of a casing and the low pressure side of the casing is provided to facilitate balance of the second rotor about the transverse axis.

31. The device as recited in claim 28 where there is an equal number of lobes for the first and second rotors.

32. The device as recited in claim 28 where each lobe of each rotor has a forward surface and a rearward surface is located between two adjacent lobes at the rearward portion of a rotor

and operating chambers are defined by the forward surface of a rotor, the adjacent rearward surface of the opposite rotor and the opposing engagement surfaces of the opposite rotor.

33. The device as recited in claim 27 where a second engagement tip of the second rotor is positioned at the forward region of the second contour surface of the second rotor and the second engagement tip of the second rotors adapted to engage a contour surface of the first rotor.

34. The device as recited in claim 5 where the housing comprises a first section and a second section where the first and second rotors and the housing comprise a first rotor section.

35. The device as recited in claim 34 where a second rotor section is placed adjacent to the first rotor section and the central shaft is connected to the first rotor of the second rotor section.

36. The device as recited in claim 35 where the casing of second section has an output port that is in communication with an input port of the a first rotor section whereby creating an in-series flow.

37. The device as recited in claim 35 where the casing of second section has an input port and an output port where the input port is in communication with an input port of the a first rotor section whereby creating an in-parallel flow of the fluid.

38. The device as recited in claim 11 where the second rotor has axial surfaces that define conduits that are provide communication between the forward portion of the second rotor and the rearward portion of the second rotor.

39. The device as recited in claim 38 where the casing has axial conduits to allow communication between the inlet port to the operating chambers.

40. The device as recited in claim 38 where the casing has axial conduits to allow communication between the outlet port to the operating chambers.

41. The device as recited in claim 39 where the casing has axial conduits to allow communication between the said outlet port and the said operating chambers.

42. The device as recited in claim 41 where the second rotor has a base surface with a positive angle that provides rotational balance about a transverse axis.

43. The device as recited in claim 41 where a conduit between the high pressure portion of a casing and the low pressure side of the casing is provided to facilitate balance of the second rotor about the transverse axis.

44. The device as recited in claim 39 where the second rotor has a base surface with a positive angle that provides rotational balance about a transverse axis.

45. The device as recited in claim 44 where a conduit between the high pressure portion of a casing and the low pressure side of the casing is provided to facilitate balance of the second rotor about the transverse axis.

46. A machine that converts energy comprising:

a housing having an inner surface;

a first rotor mounted for rotation in the housing about a first axis, a first outer surface defining at least part of a sphere having a common center with the first inner surface and adapted to intimately engage the inner surface of the housing;

a second rotor having a forward portion and a rearward portion, mounted for rotation on the housing about a second axis offset from the first axis and being collinear by an angle α and intersecting at the common centers of the rotors, a second outer surface defining at least part of a sphere and having a common center with the second inner surface and adapted to engage the inner surface of the housing;

the said first rotor further having a first contour face that is defined by the locus formed by points on the second rotor as the second rotor rotates about the second axis, and a first contour surface is positioned in the forward region of the first rotor;

the second rotor further having a second contour face that is defined by the locus formed by points on the first rotor as the first rotor rotates about the first axis, the second rotor further having a rearward surface that is positioned in the rearward portion of the second rotor;

the points of each rotor that define the locus line along and outer edges of a common central axis is essentially a radius extending outward from the common centers of the rotor at an angle $\alpha/2$ from a normal to the axis of the other rotor;

whereas the first contour surface of the first rotor does not come into contour with the rearward surface of the second rotor allowing fluid to pass thereinbetween.

47. The apparatus as recited in claim 46 further comprising:

whereas the machine to convert energy is a pump that is adapted to increase the pressure of a fluid, and the

housing has a first lateral radial portion and a second has an input port located on and an output port

48. An assembly adapted to increase the pressure of a fluid where the assembly comprises:

a central shaft having a longitudinal central axis and is adapted to rotate about said central axis, said central shaft further having at least part of a cylindrical outer surface;

a rotor assembly comprising;

a power rotor adapted to rotate about a longitudinal power axis the power rotor comprising, a first outer surface defining at least part of a sphere having a common center with the first inner surface and adapted to intimately engages the inner surface of the housing, the power rotor having an inward region and an outward region and comprising a plurality of lobes and further comprising an outward contour surface and longitudinally extending surfaces defining conduits allowing communication between the inward region and the outward region, a first contour surface that is positioned in the forward region of the first rotor;

a slave rotor adapted to rotate about a longitudinal slave axis and having an inward region and an outward region;

a base housing having a central portion and a peripheral portion, the base housing further having a master region and a slave region the base housing comprising;

a central surface located in the central portion and is adapted to be in close engagement of the cylindrical surface of the central shaft;

a first surface adapted to engage the outward surface of the said power rotor,

a first longitudinally extending surface defining a first passageway allowing communication to the power conduits of the power rotor,

a second surface located in the slave region of the base housing and is adapted to engage the outward surface of the slave rotor and support the slave rotor about the longitudinal slave axis at an angle α with respects to the longitudinal power axis,

whereas the said first rotor further having a first contour face that is defined by the locus formed by points on the second rotor as the second rotor rotates about the second axis, the second rotor further having a first contour face that is defined by the locus formed by points on the first rotor as the first rotor rotates about the first axis, and the central surface of the power rotor is connected to the central shaft and the inward regions of the slave rotor and the power rotor are adapted to engage one another and rotate where the lobes of the slave rotor and the power rotor define operating chambers that change in volume with respects to rotation of the central shaft and fluid is displaced through the

conduits of the power rotor and through the first passageway.

49. The device as recited in claim 48 where the second rotor is substantially rotationally balanced about the second axis with respects to the rotational position of the second rotor.

50. The device as recited in claim 48 where the first rotor has a center surface defining the said inner component and is adapted to engage the second inner surface of the second rotor.

51. The device as recited in claim 48 where the first and second rotors have inward surfaces adapted to allow the said shaft to pass therethrough.

52. The device as recited in claim 51 where a shaft extends through the inward surface of the first rotor and is attached thereto.

53. The device as recited in claim 52 where a torque is applied to the said shaft and fluid is compressed in operating chambers defined in part by the contour faces of the first and second rotors.

54. The device as recited in claim 52 where the inner surface of the first rotor engages the shaft in a manner to rotate in conjunction therewith.

55. The device as recited in claim 54 where the second rotor is substantially balanced about the second axis with respects to the rotational position of the second rotor.

56. The device as recited in claim 46 where the inlet port and outlet ports are positioned on respective opposite sides of a transverse axis of the rotors.

57. The device as recited in claim 56 where the casing engages the radially outer surface of the rotors at bottom dead center and top dead center.

58. The device as recited in claim 57 where the operating chambers while located at the high pressure port are adapted to increase the pressure of a fluid contained therein.

59. The device as recited in claim 58 where the fluid is incompressible.

60. The device as recited in claim 58 where the fluid is a compressible gas.

61. The device as recited in claim 46 where there is an equal number of lobes for the first and second rotors.

62. The device as recited in claim 61 where each lobe of each rotor has a forward surface and a rearward surface is located between two adjacent lobes at the rearward portion of a rotor and operating chambers are defined by the forward surface of a rotor, the adjacent rearward surface of the opposite rotor and the opposing engagement surfaces of the opposite rotor.

63. The device as recited in claim 46 where the rotor assembly and the base housing comprise a first rotor section.

64. The device as recited in claim 63 where a second rotor section is placed adjacent to the first rotor section and the central shaft is connected to the first rotor of the second rotor section.

65. The device as recited in claim 64 where the casing of second section has an output port that is in communication with an input port of the a first rotor section whereby creating an in-series flow.

66. The device as recited in claim 65 where the casing of second section has an input port and an output port where the input port is in communication with an input port of the a first rotor section whereby creating an in-parallel flow of the fluid.

67. The device as recited in claim 66 where a conduit between the high pressure portion of a casing and the low pressure side of the casing is provided to facilitate balance of the second rotor about the transverse axis.

68. A method of displacing a fluid comprising:

retrieving a housing having an inner surface defining at least part of a sphere and positioning a first rotor mounted for rotation in the housing about a first axis and having a forward region and a rearward region and a first outer surface defining at least part of a sphere and adapted to intimately engages the inner surface of the housing and positioning an inner component have an outer surface that defines at least part of a sphere at the center of the inner surface of the housing;

positioning a second rotor in the said housing having a forward portion and a rearward portion, mounted for rotation in the housing about a second axis offset from being collinear with the first axis by an angle α and intersecting at the common centers of the rotors, the second rotor including a second inner surface defining at least part of a sphere having a common center with the center of the first rotor and is adapted to engage the said inner component, a second outer surface defining at least part of a sphere and having a common center with the

second inner surface and adapted to engage the inner surface of the housing;

whereas the said first rotor further having a first contour surface that is defined by a locus formed by points on the second rotor as the second rotor rotates about the second axis, and the first rotor further has a first engagement tip which is positioned in the forward region of the first rotor, the second rotor further having a second engagement surface that is defined by a locus formed by points on the first rotor as the first rotor rotates about the first axis, the second engagement surface having a base region and the points of each rotor that define the locus lie along an outer edge of a central axis is essentially a radius extending outward from the common centers of the rotor at an angle $\alpha/2$ from a normal to the axis of the other rotor and the shaft extends through the first and second rotors where rotation of the shaft changes the volume of operating chambers formed between the contour faces of the first and second rotors.

69. The method as recited in claim 68 whereas the first engagement tip of the first rotor does not come into contact with the base region of the second rotor allowing fluid to communicate thereinbetween.

70. The method as recited in claim 68 where the second rotor has axial surfaces that define conduits that are provide communication between the forward portion of the second rotor and the rearward portion of the second rotor.

71. The method as recited in claim 70 where the casing has axial conduits to allow communication between the inlet port to the operating chambers.

72. The method as recited in claim 70 where the casing has axial conduits to allow communication between the outlet port to the operating chambers.

73. The method as recited in claim 71 where the casing has axial conduits to allow communication between the said outlet port and the said operating chambers.

74. The method as recited in claim 72 where the second rotor has a base surface with a positive angle that provides rotational balance about a transverse axis.

75. The method as recited in claim 72 where a conduit between the high pressure portion of a casing and the low pressure side of the casing is provided to facilitate balance of the second rotor about the transverse axis.

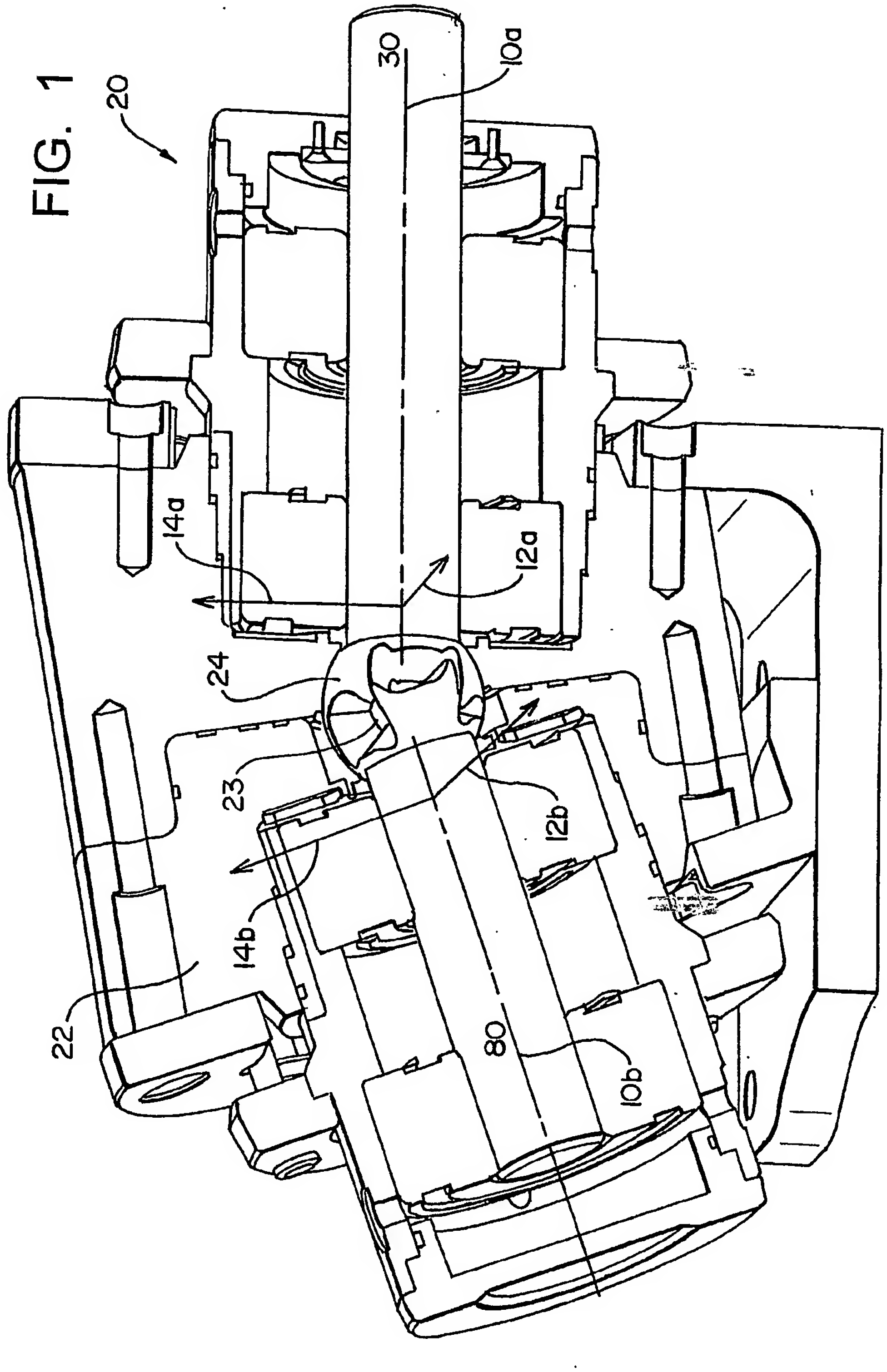


FIG. 1

FIG. 1A

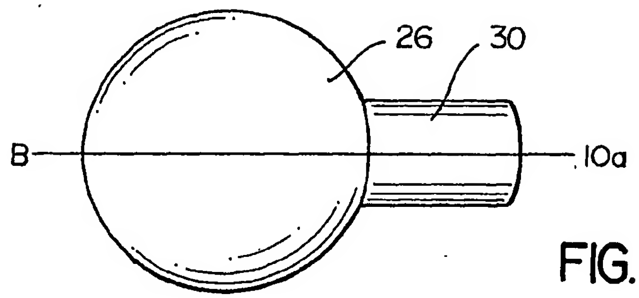


FIG. 1B

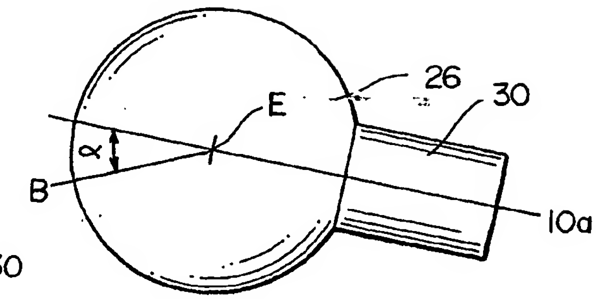


FIG. 1C

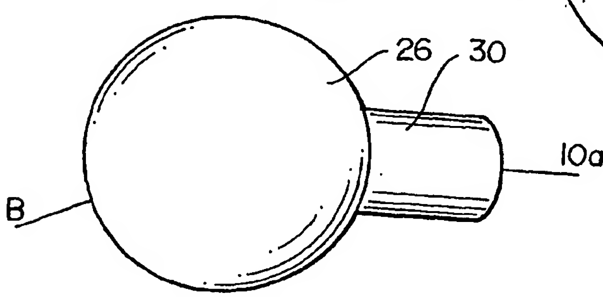


FIG. 2A

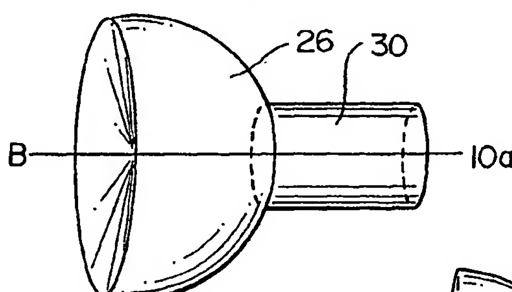


FIG. 2B

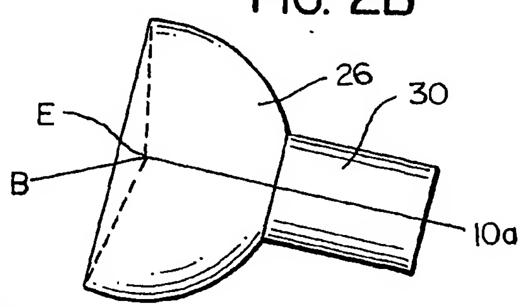


FIG. 2C

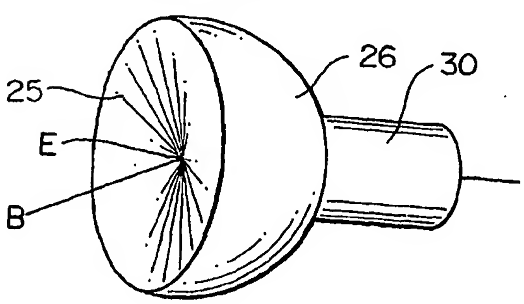


FIG. 3A

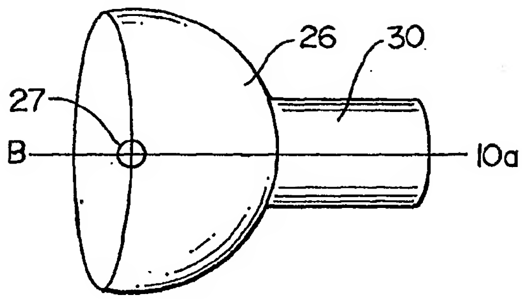


FIG. 3B

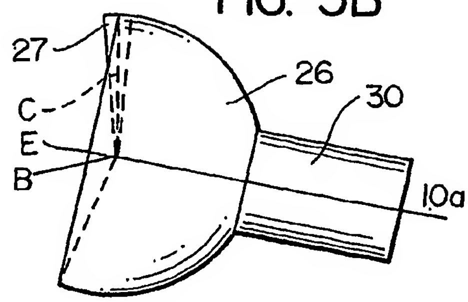


FIG. 3C

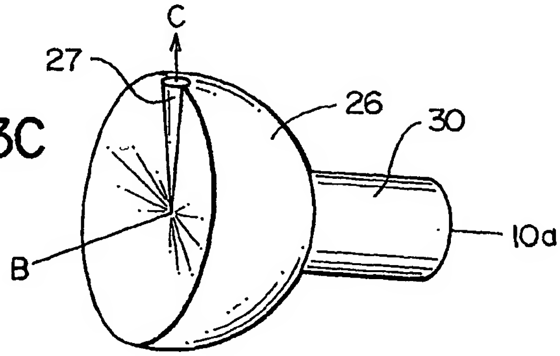


FIG. 4A

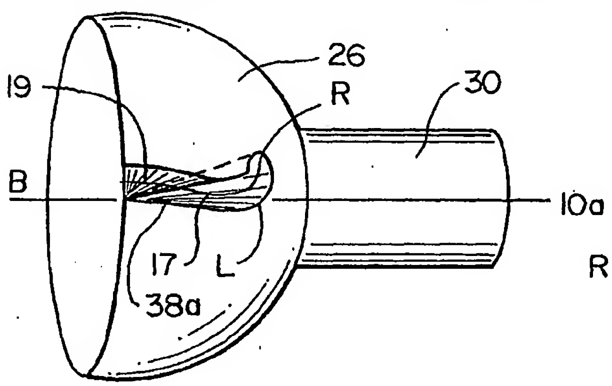


FIG. 4B

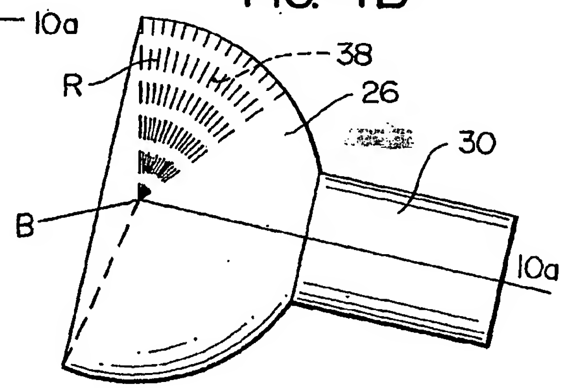


FIG. 4C

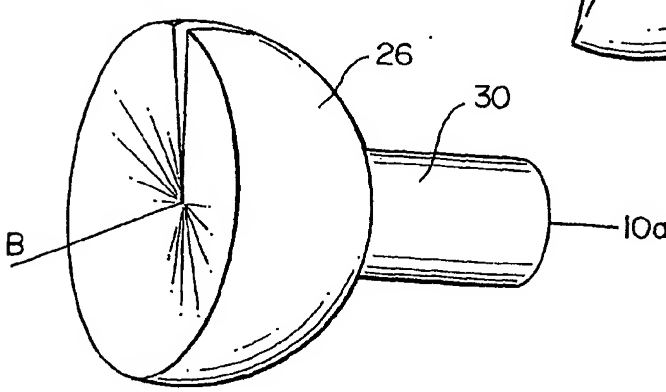


FIG. 4D

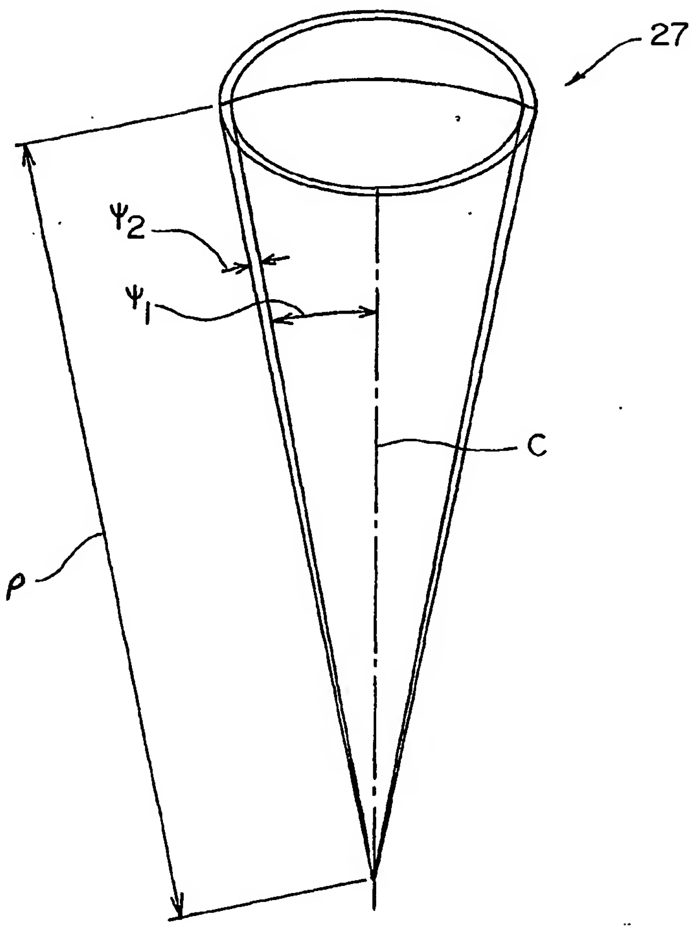


FIG. 5B

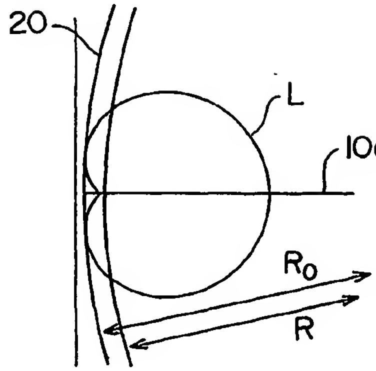


FIG. 5A

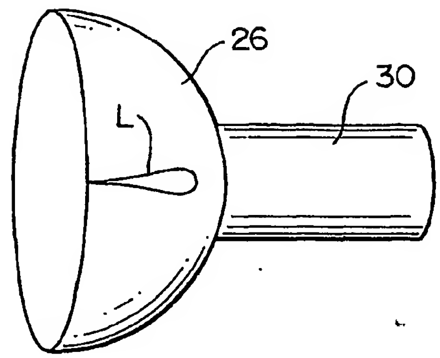


FIG. 6A

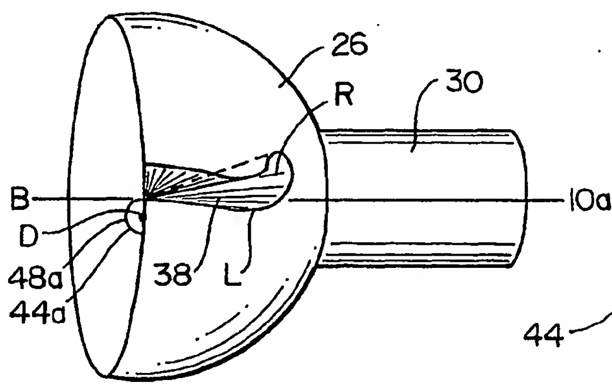


FIG. 6B

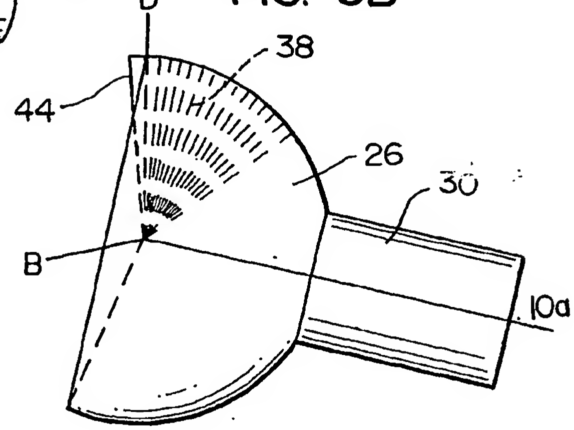
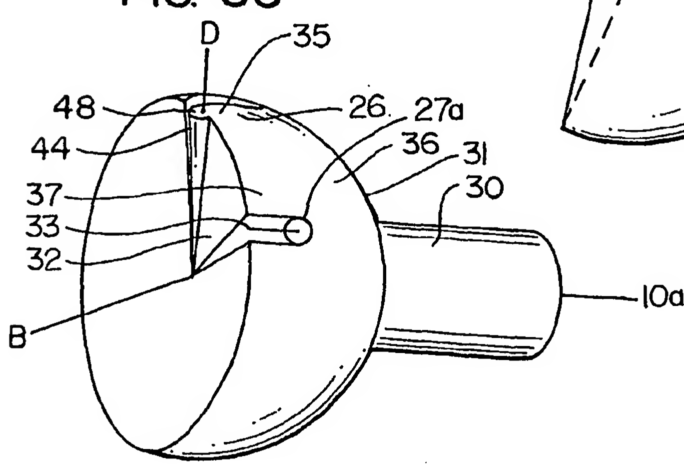


FIG. 6C



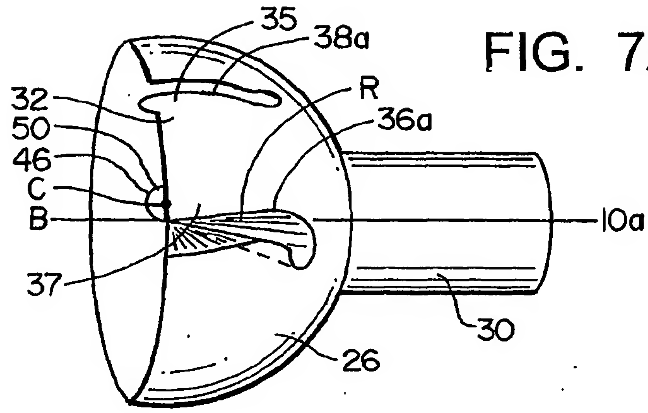


FIG. 7A

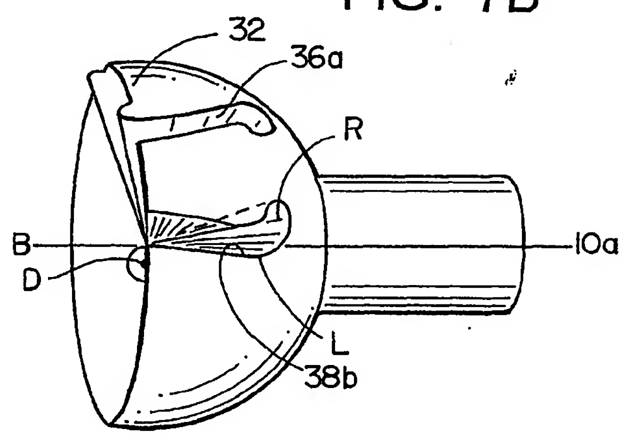


FIG. 7B

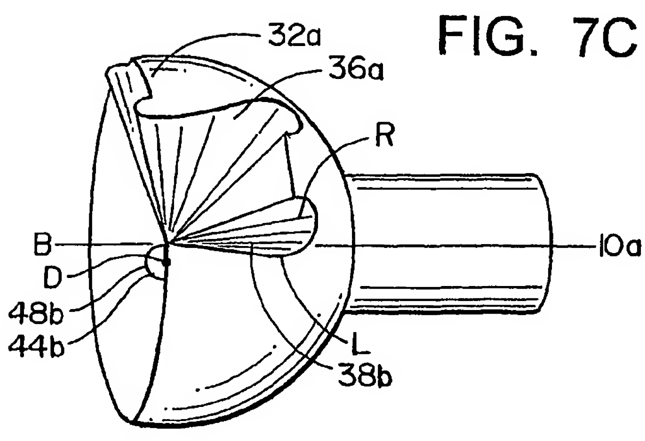


FIG. 7C

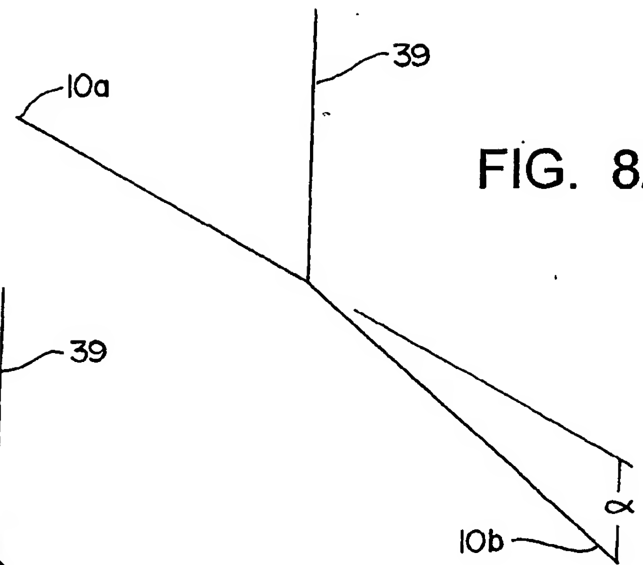


FIG. 8A

FIG. 8B

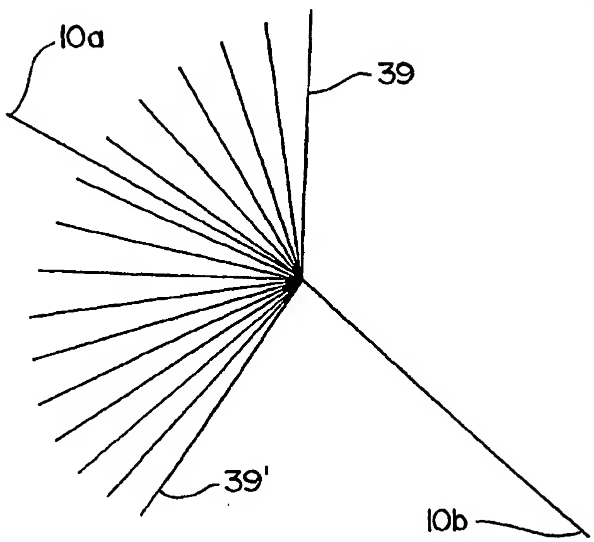


FIG. 8C

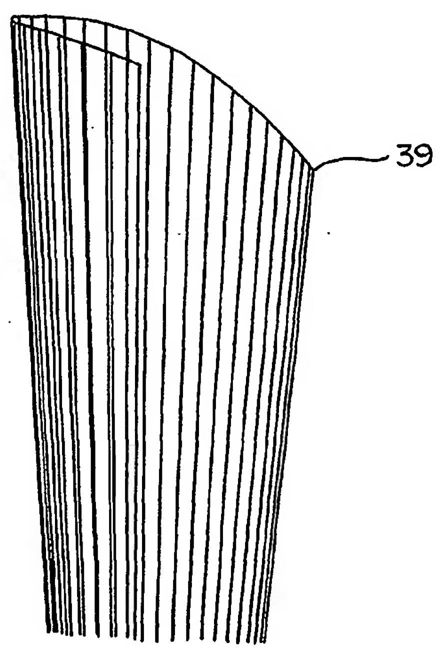


FIG. 8D

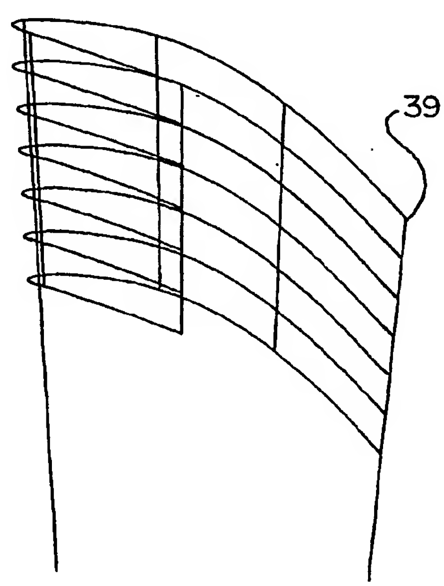


FIG. 8E

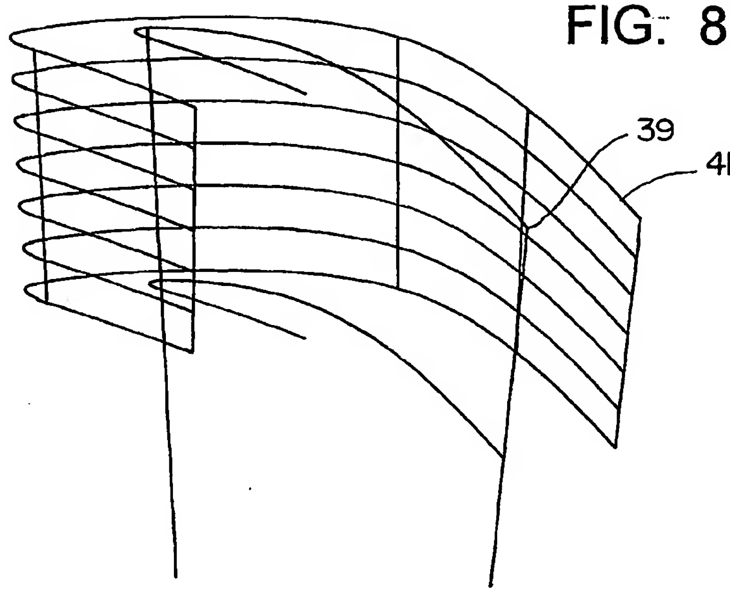


FIG. 8F

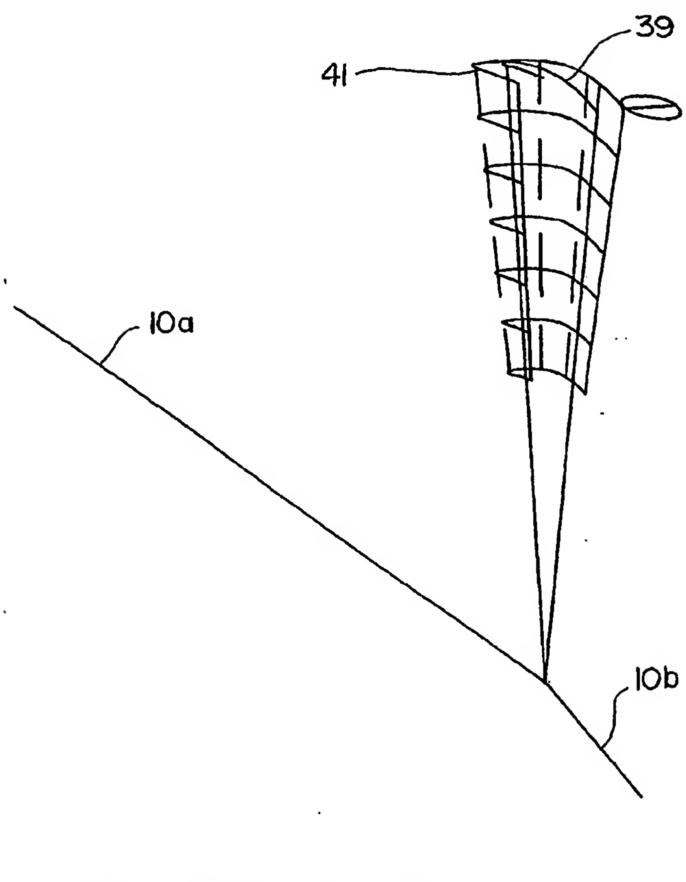


FIG. 9

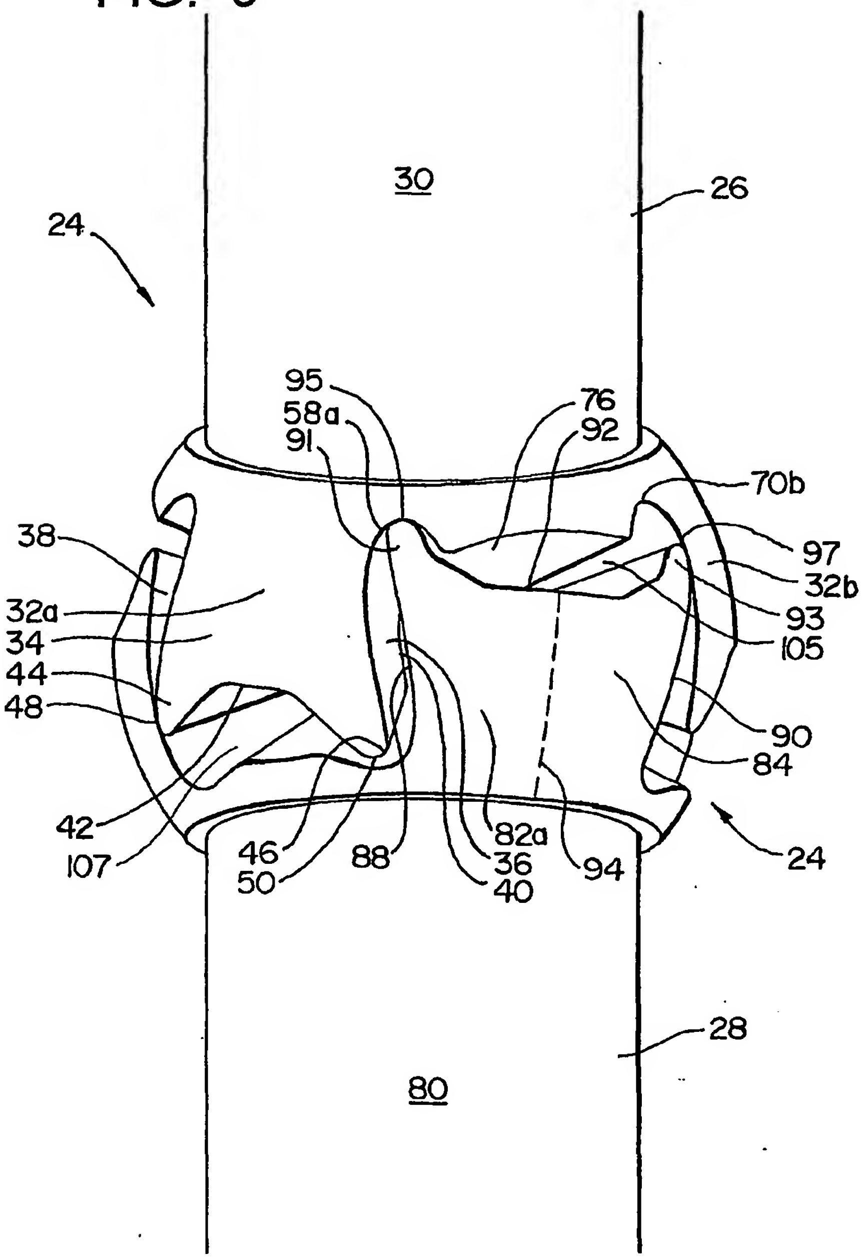

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FIG. 10A

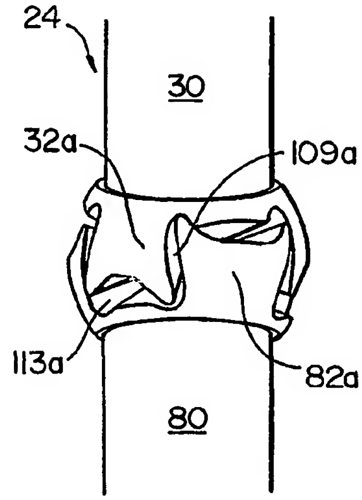

FIG. 10B

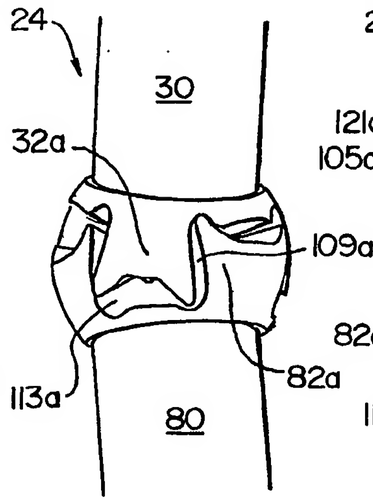

FIG. 10C

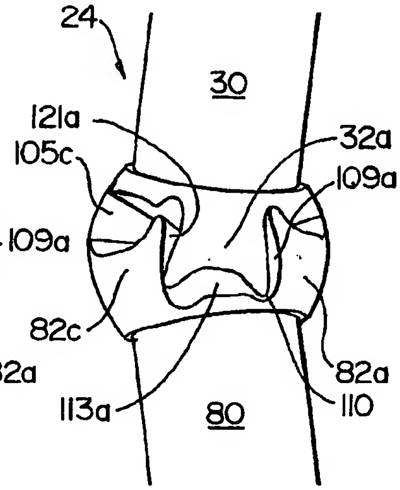

FIG. 10D

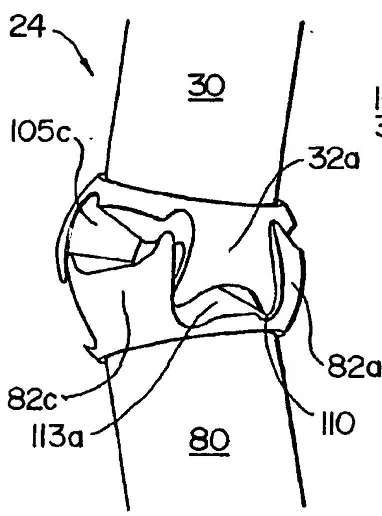

FIG. 10E

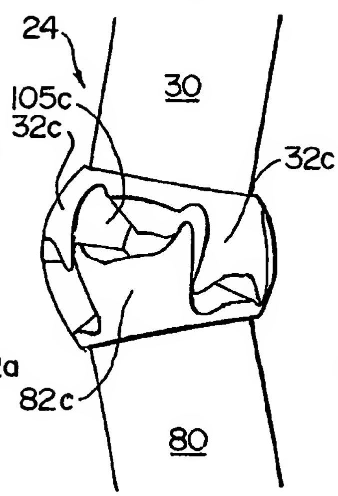

FIG. 10F

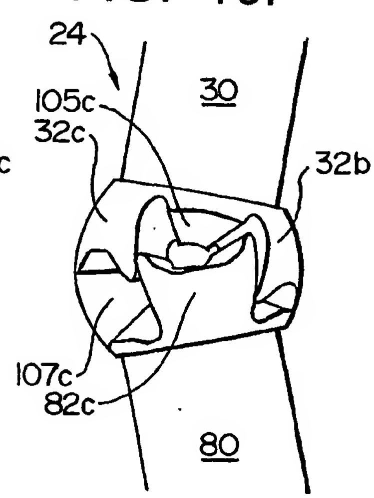

FIG. 10G

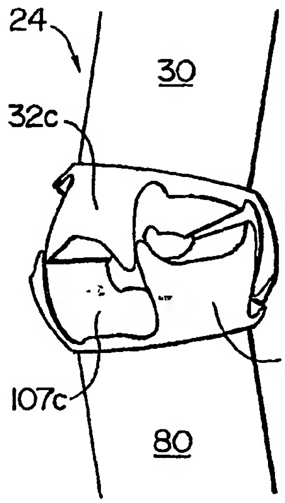

FIG. 10H

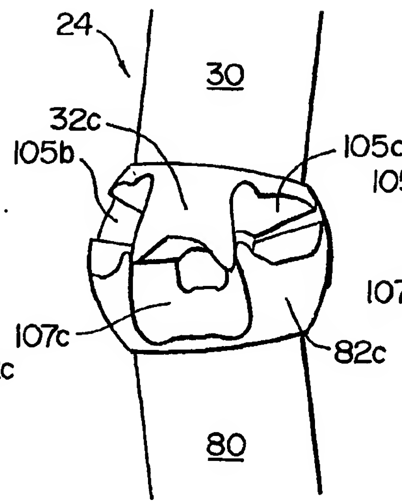

FIG. 10I

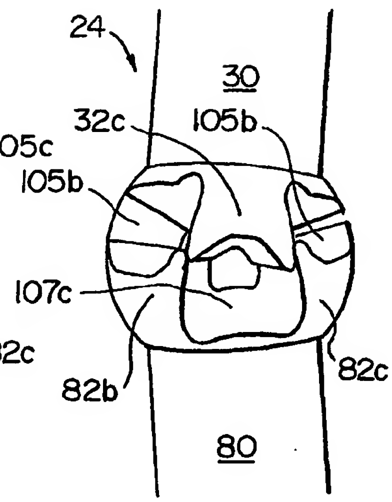

FIG. 10J

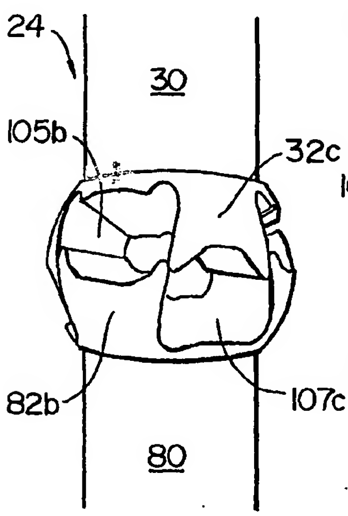

FIG. 10K

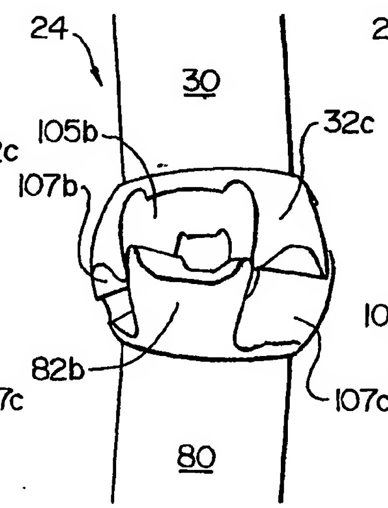

FIG. 10L

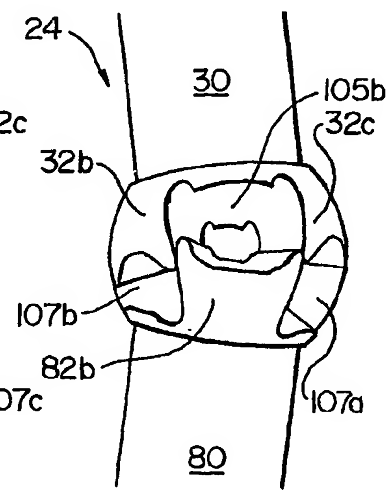

FIG. 10M

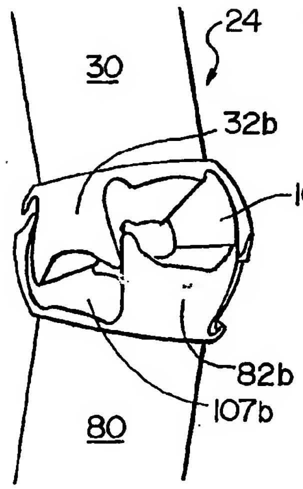

FIG. 10N

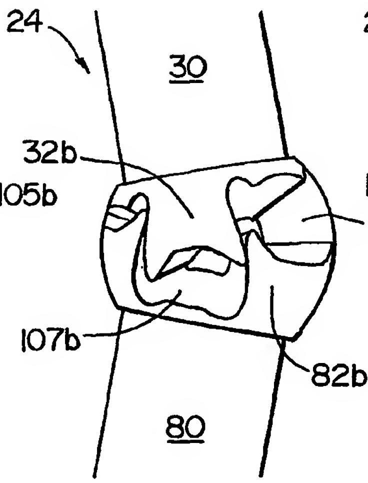

FIG. 10O

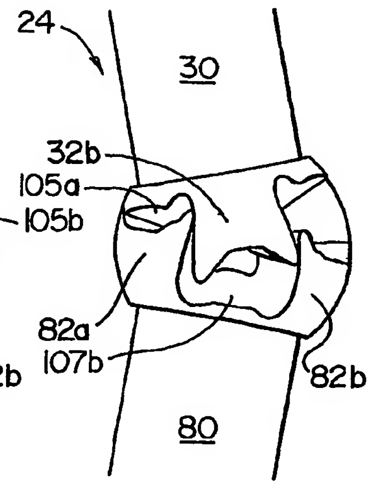

FIG. 10P

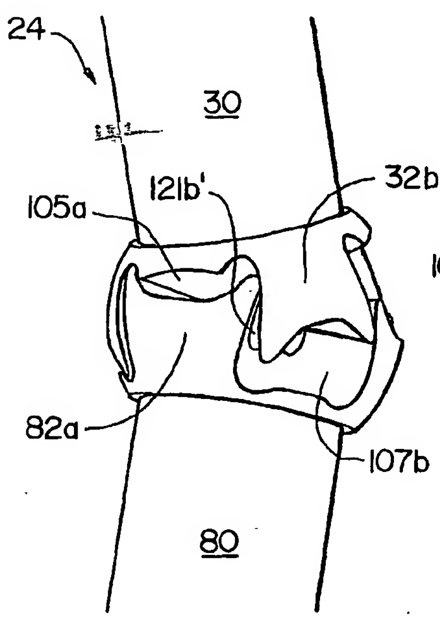

FIG. 10Q

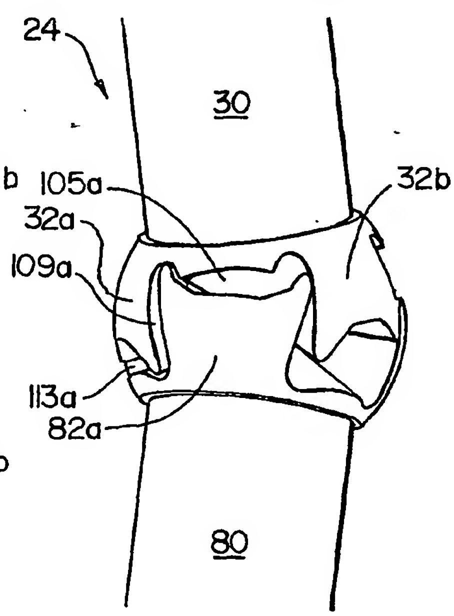

FIG. 10R

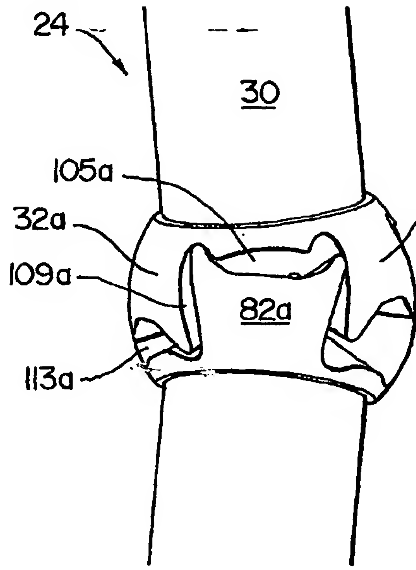

FIG. 10S

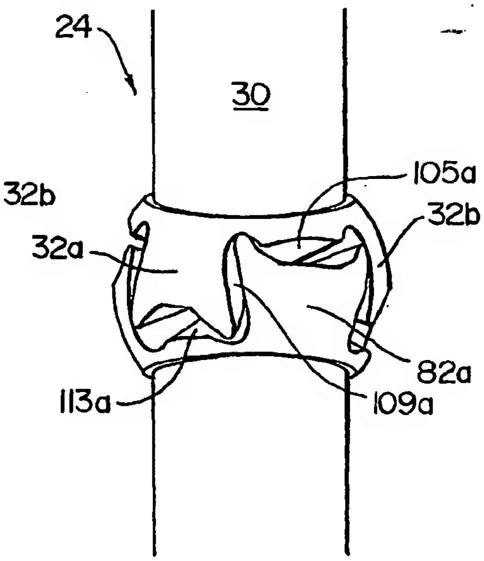

FIG. 11

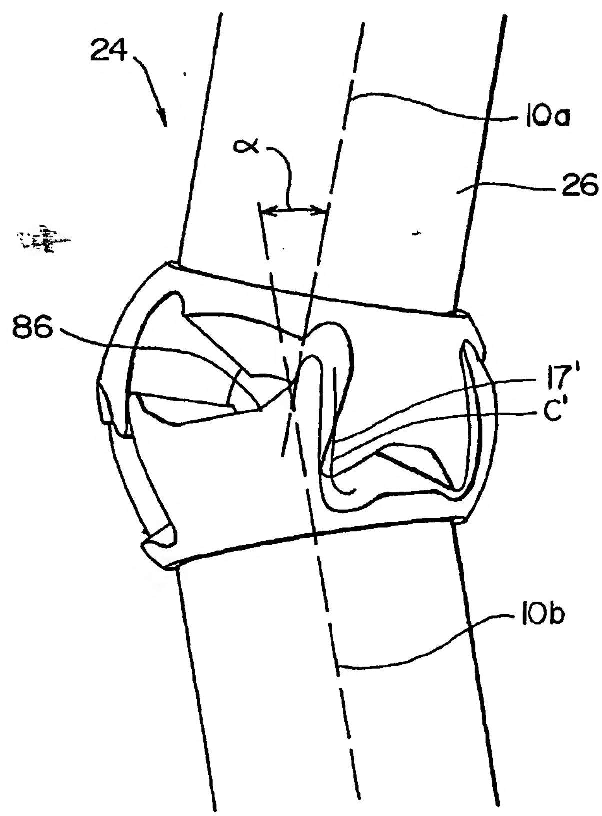

FIG. 12

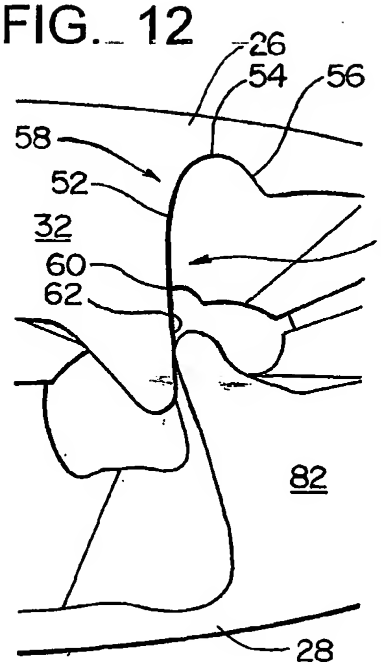

FIG. 13

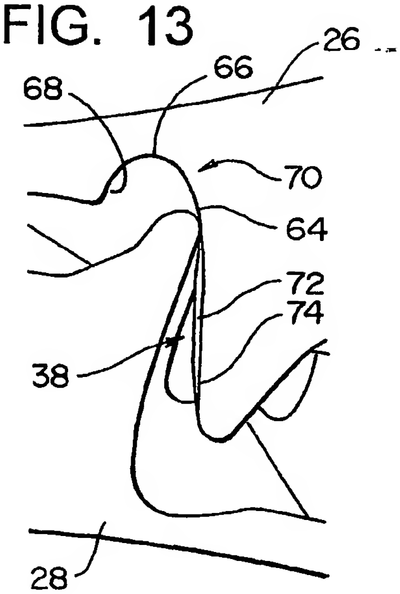

FIG. 14

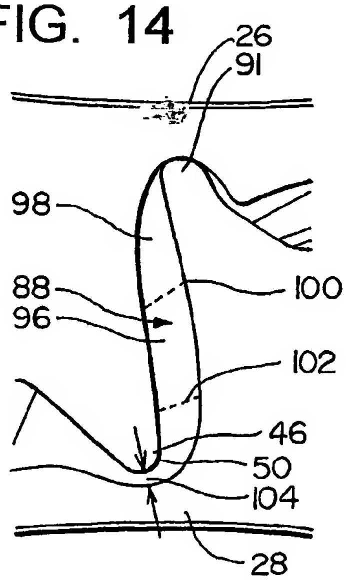

FIG. 15

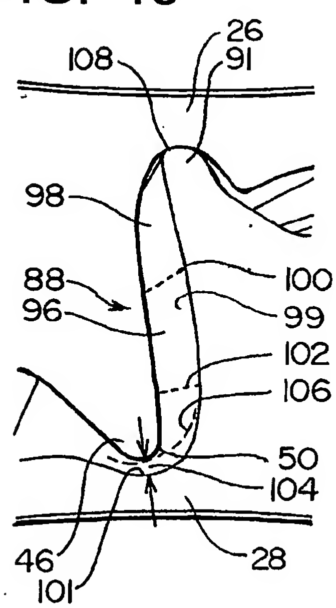

FIG. 16

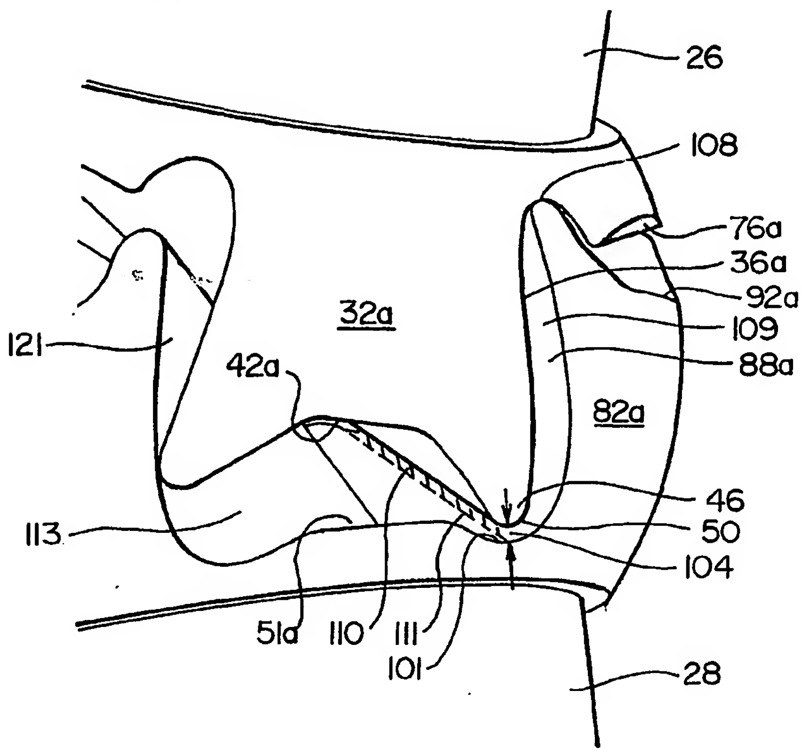
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FIG. 17A

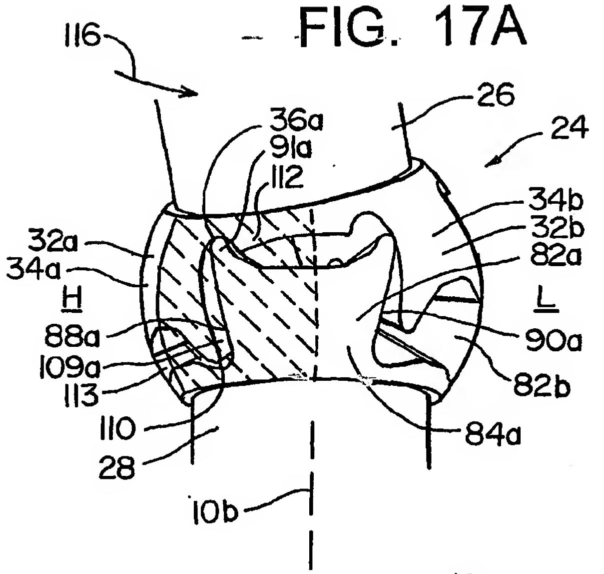


FIG. 17B

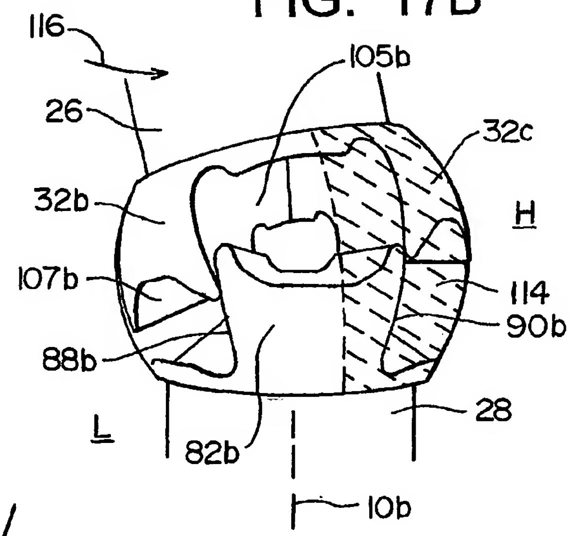


FIG. 17C

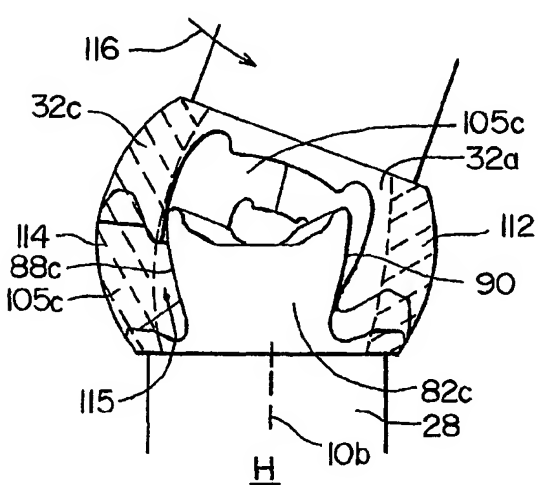


FIG. 18A

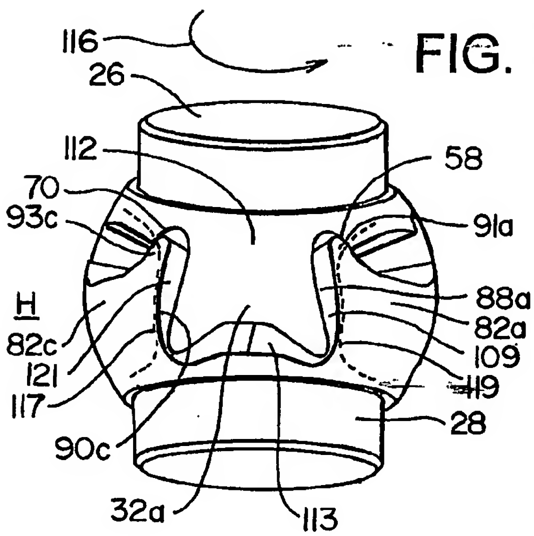


FIG. 18B

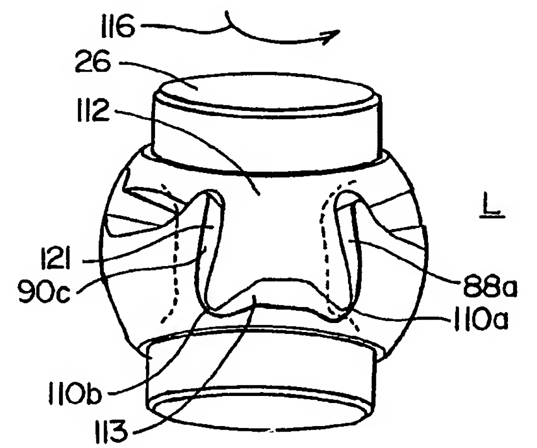


FIG. 18C

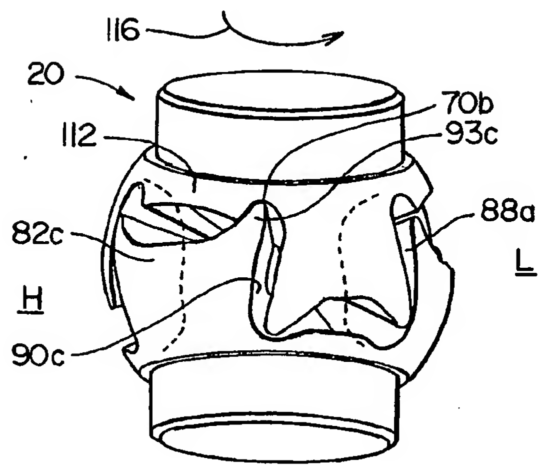


FIG. 18D

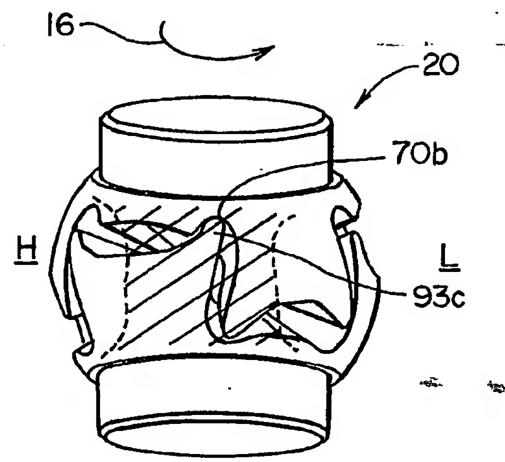


FIG. 18E

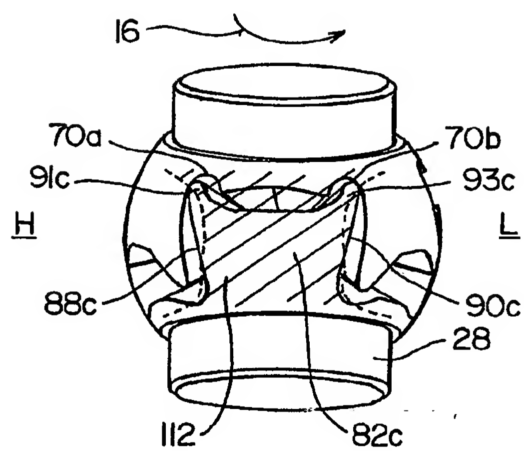


FIG. 18F

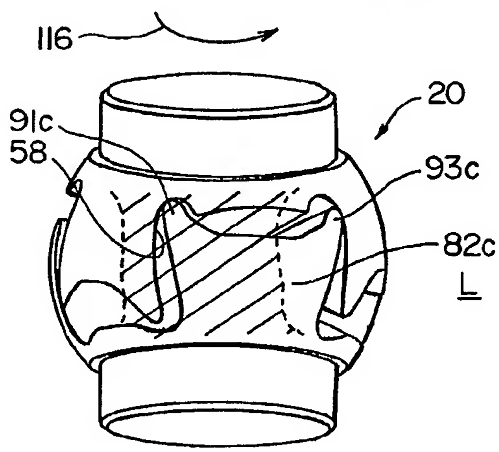


FIG. 19

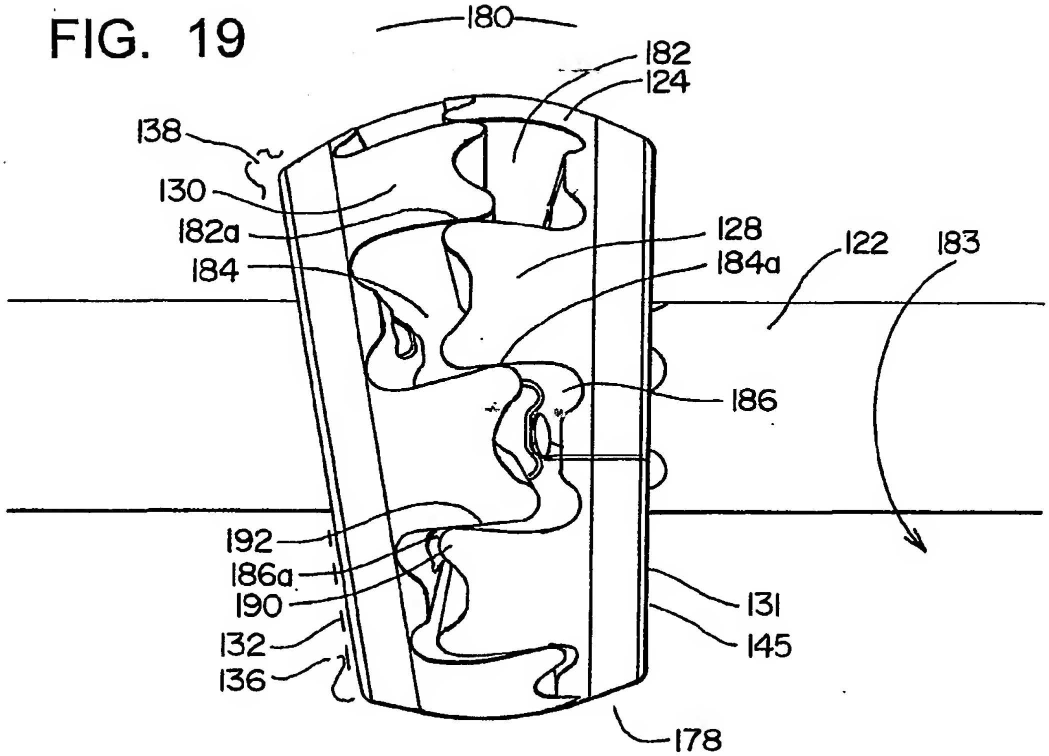

180
182
124
138
130
182a
128
184
184a
122
183
186
192
186a
190
131
132
145
136
178

FIG. 20

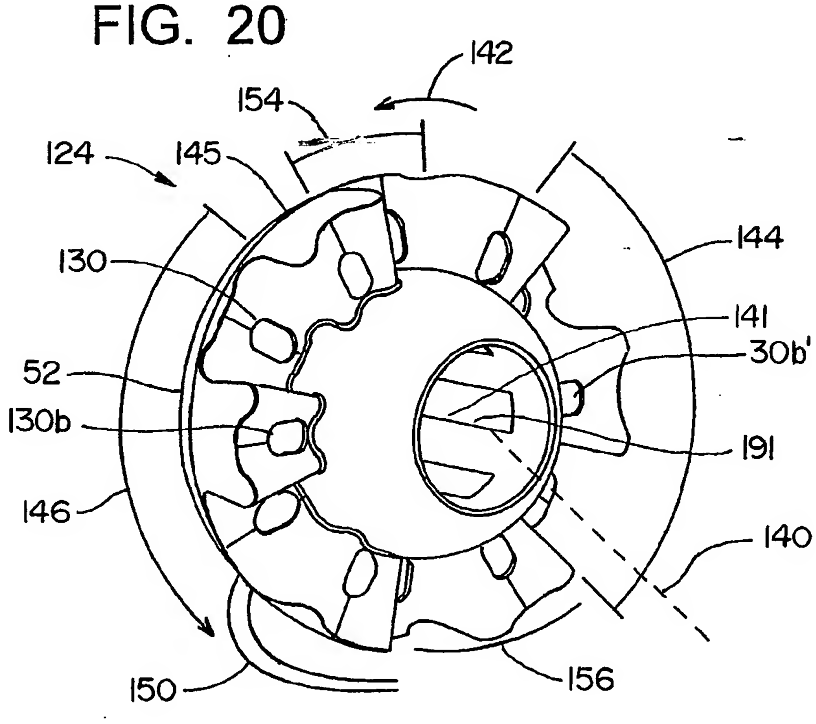

142
154
124
145
144
130
141
30b'
52
130b
191
146
140
150
156

FIG. 21

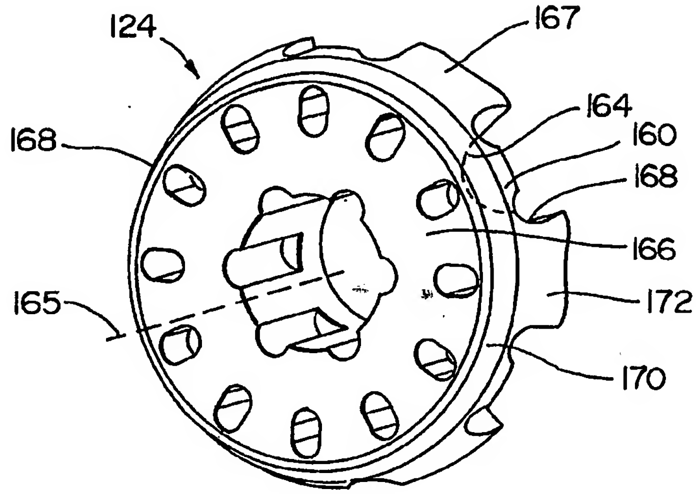
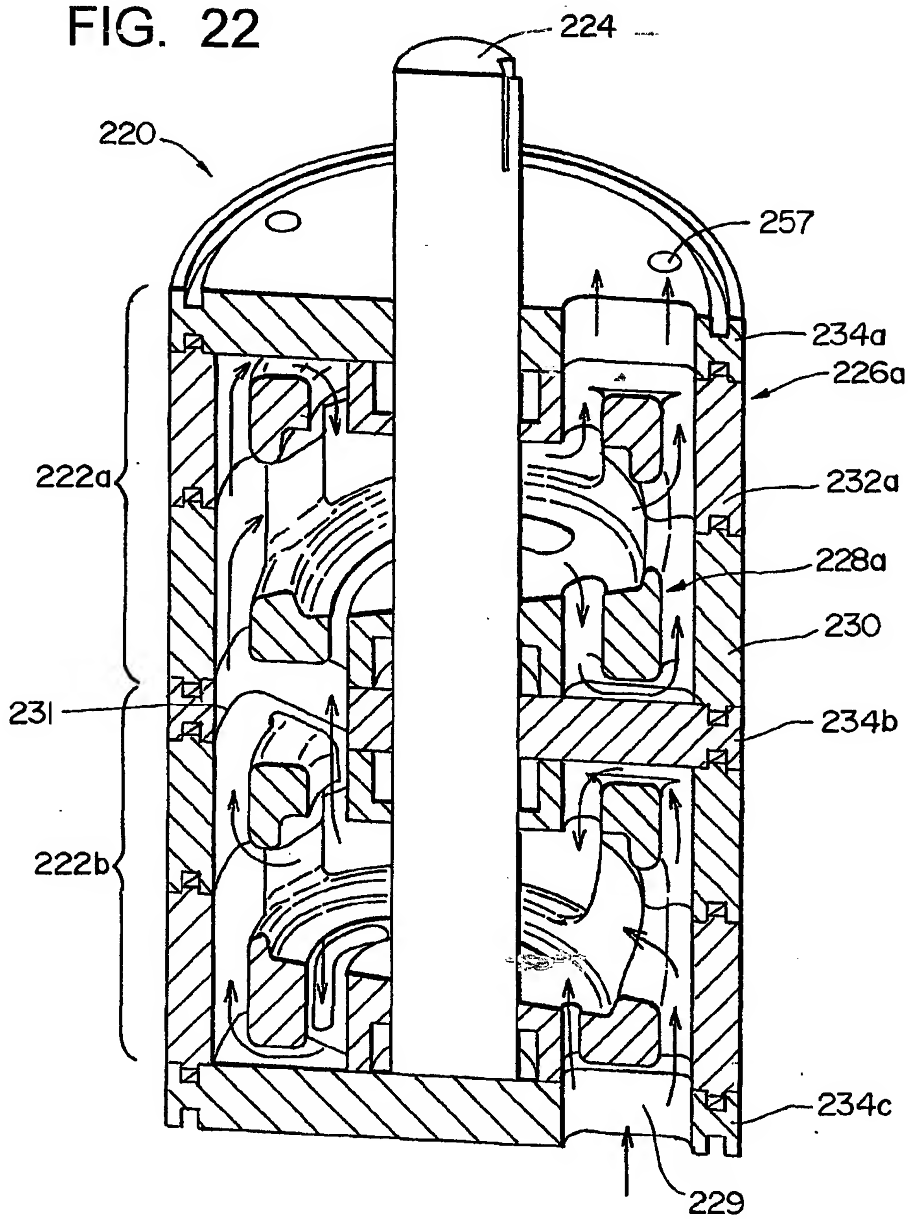


FIG. 22


224
220
257
234a
226a
222a
232a
228a
230
231
234b
222b
234c
229

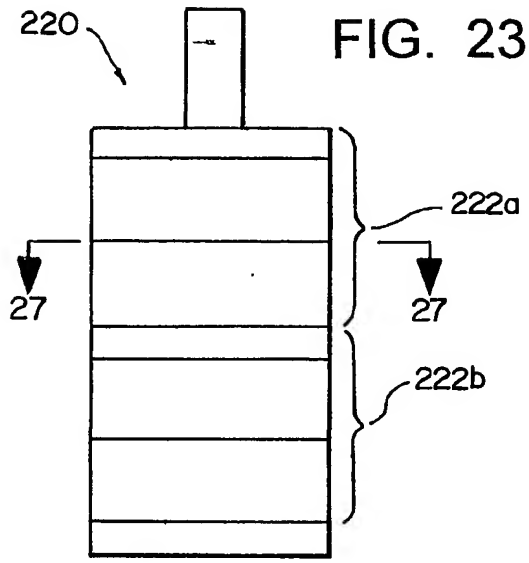

220
222a
27
27
222b

FIG. 23

FIG. 24

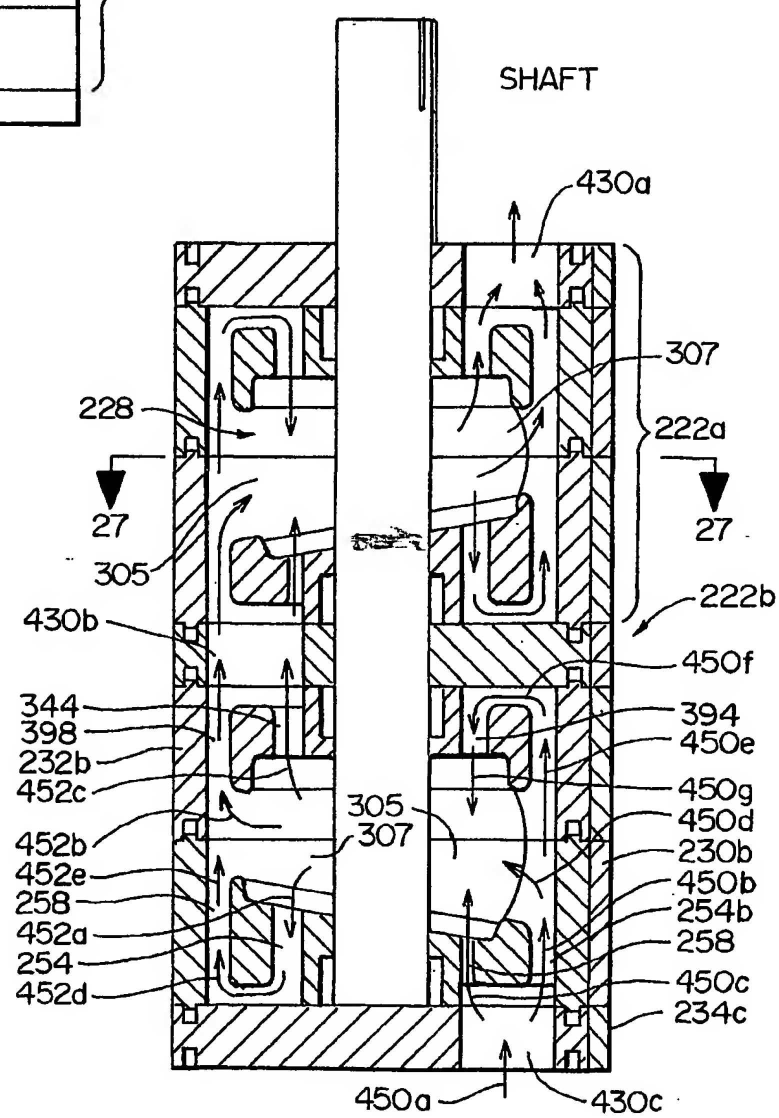

SHAFT
430a
307
228
222a
27
27
305
222b
430b
450f
344
398
232b
452c
394
450e
450g
305
307
452b
450d
452e
230b
258
450b
452a
254b
254
258
452d
450c
234c
450a
430c

FIG. 25

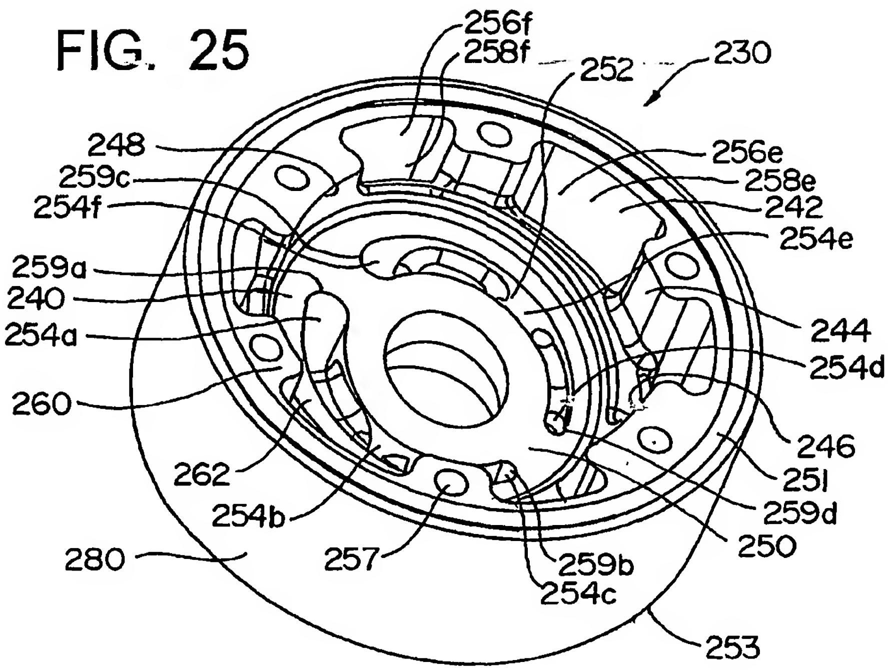


FIG. 26A

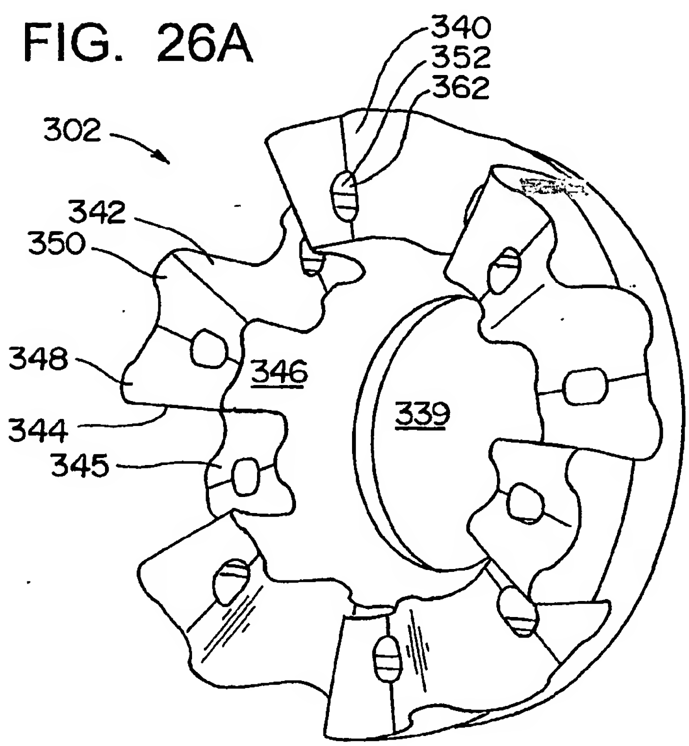


FIG. 26B

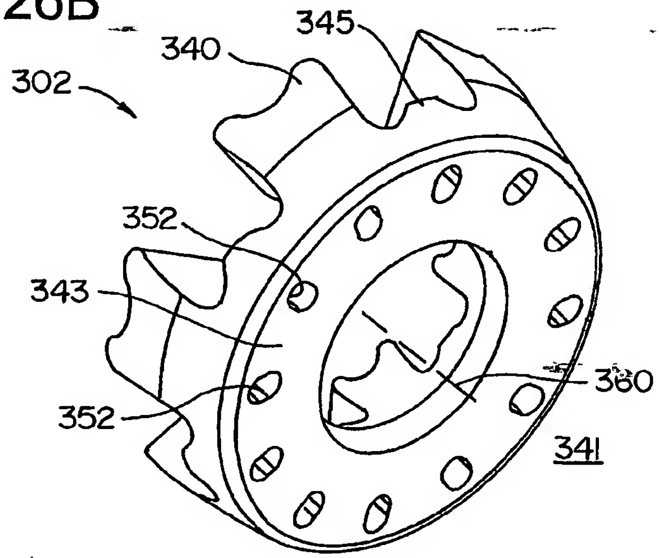


FIG. 27

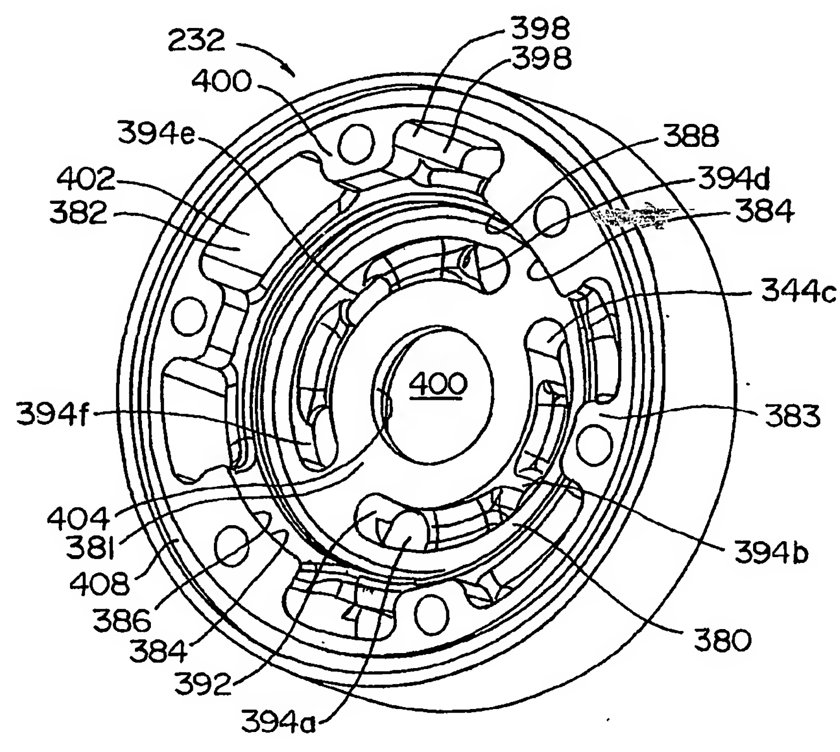


FIG. 28

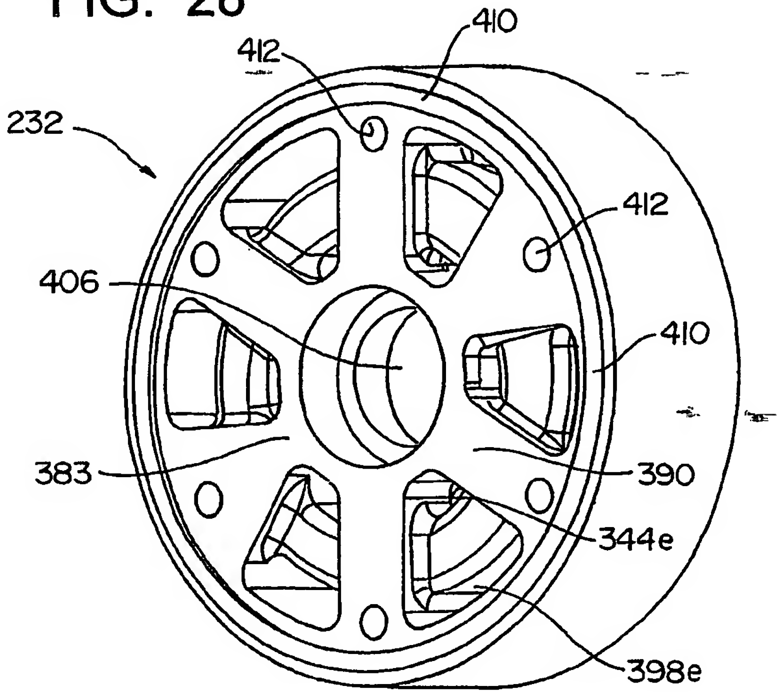

410
412
232
412
406
410
383
390
344e
398e

FIG. 29

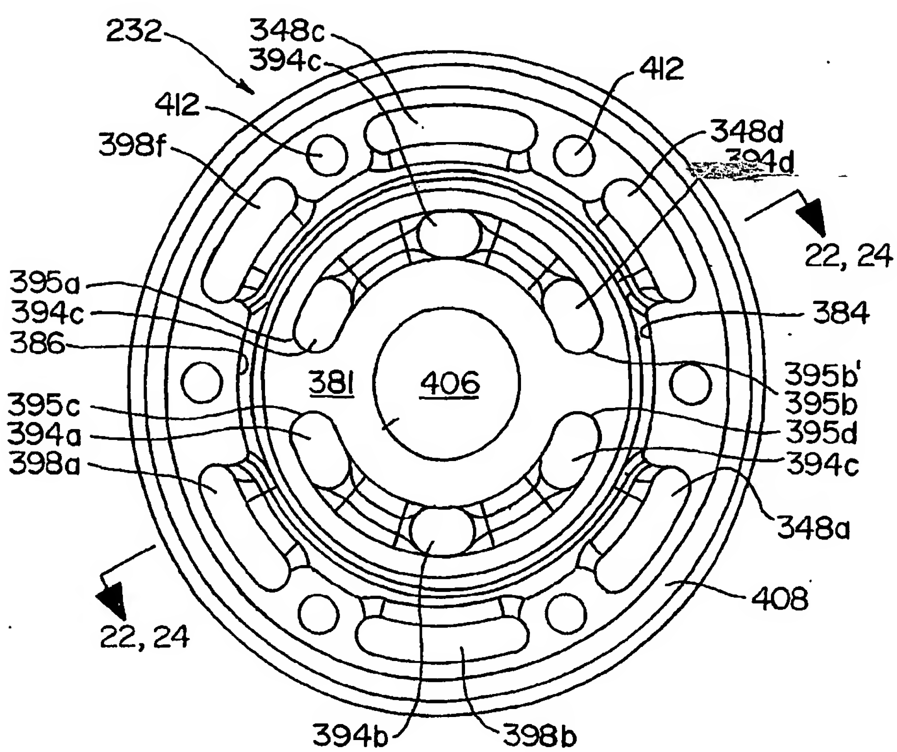

232
348c
394c
412
412
398f
348d
394d
22, 24
395a
394c
386
384
381
406
395b'
395b
395c
394a
398a
395d
394c
348a
408
22, 24
394b
398b

FIG. 30

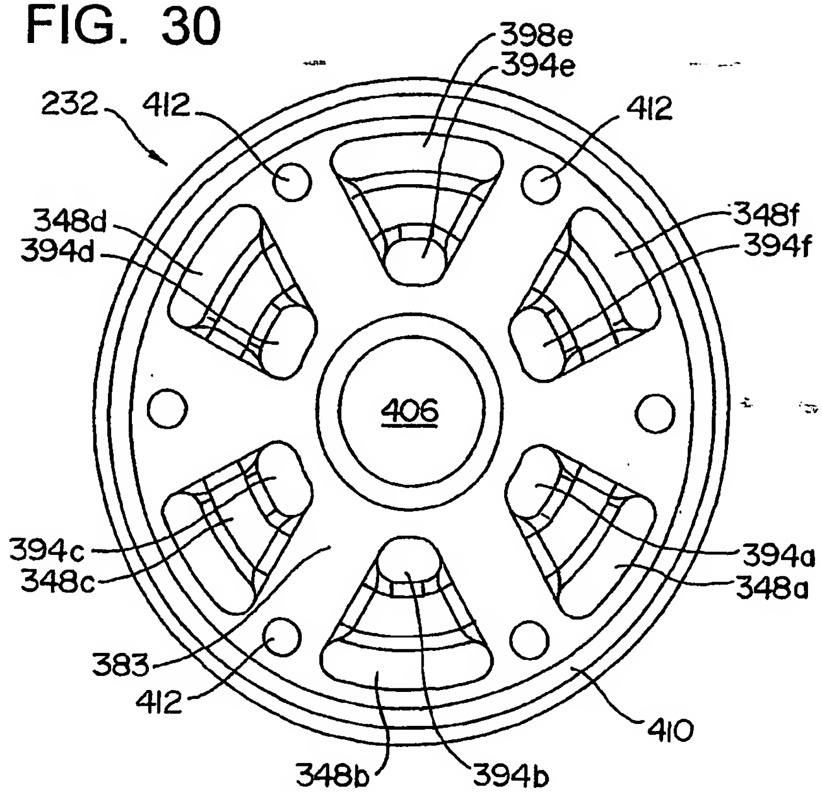

398e
394e
232
412
412
348d
394d
348f
394f
406
394c
348c
394a
348a
383
412
410
348b
394b

FIG. 31

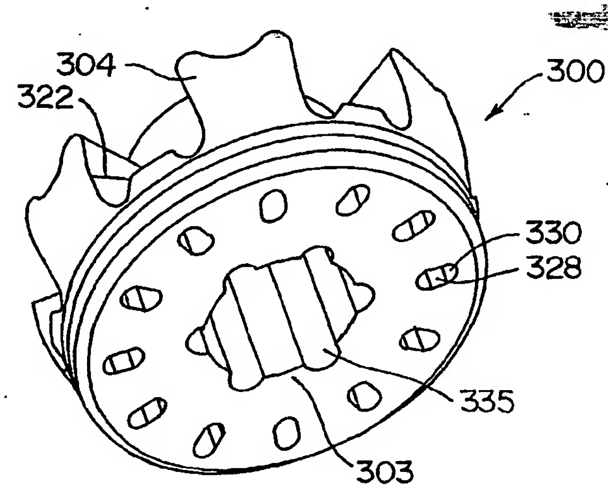

304
322
300
330
328
335
303

FIG. 32

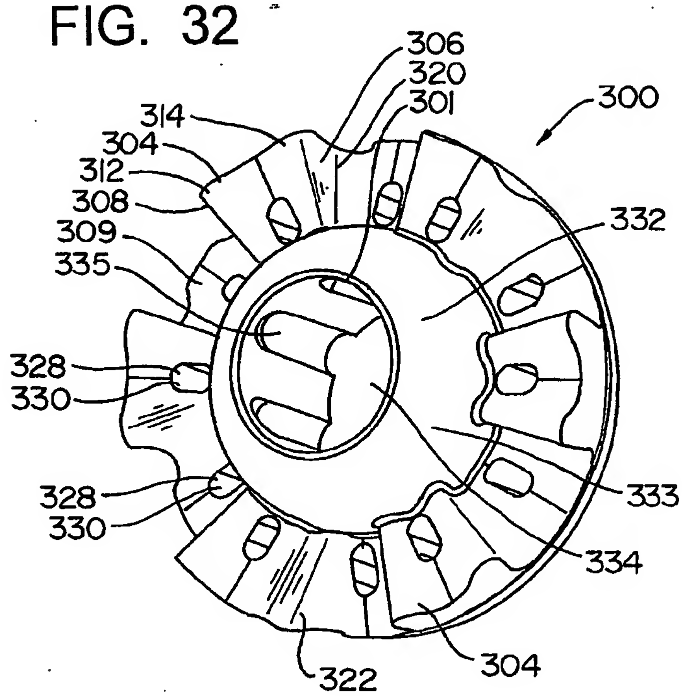

306
320
301
300
314
304
312
308
309
335
332
328
330
328
330
333
334
322
304

FIG. 33

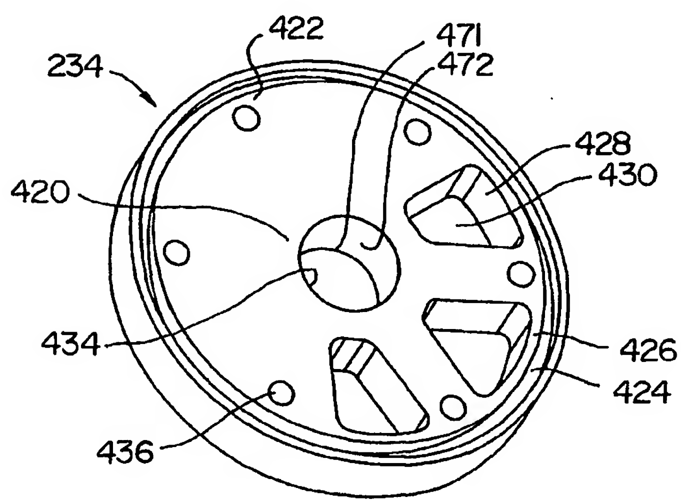

422
471
472
234
428
430
420
434
426
424
436

FIG. 34

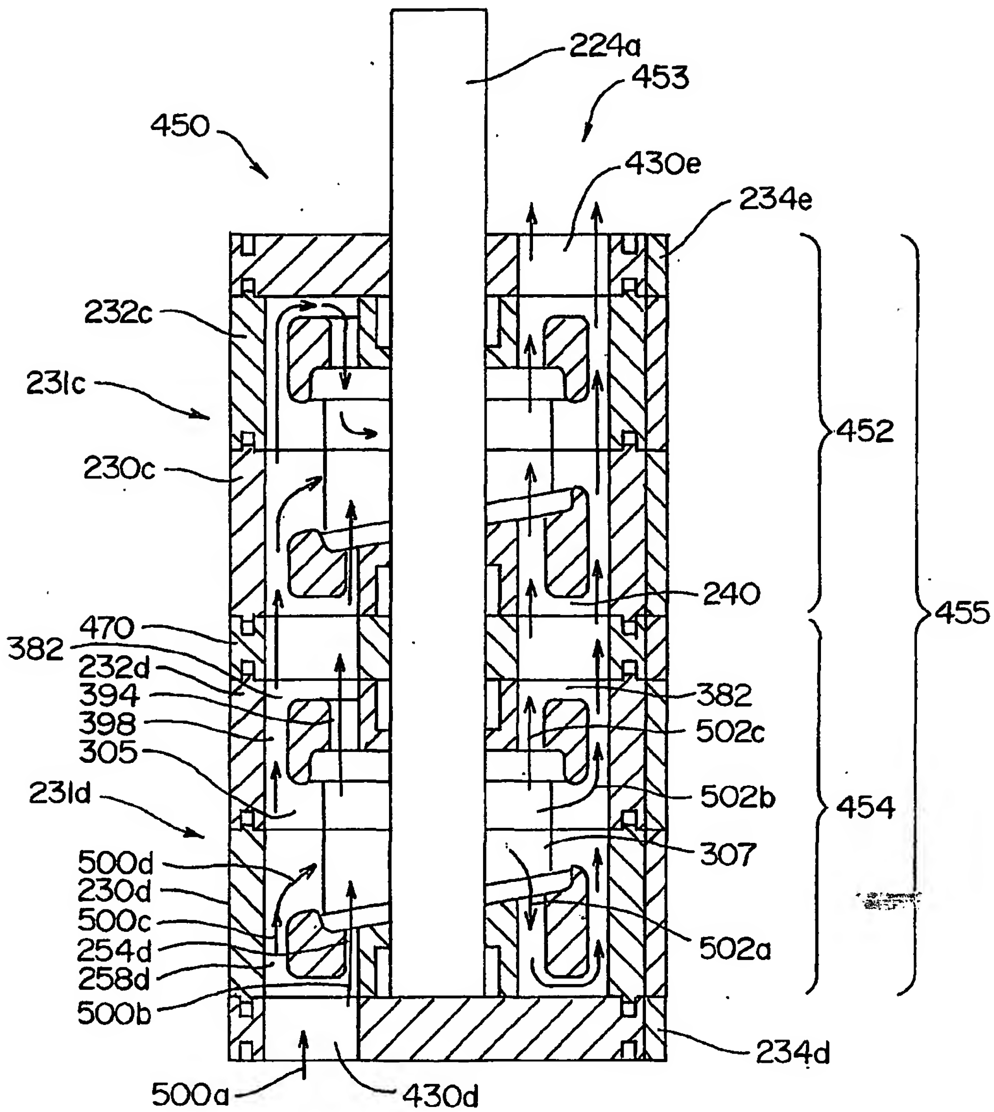
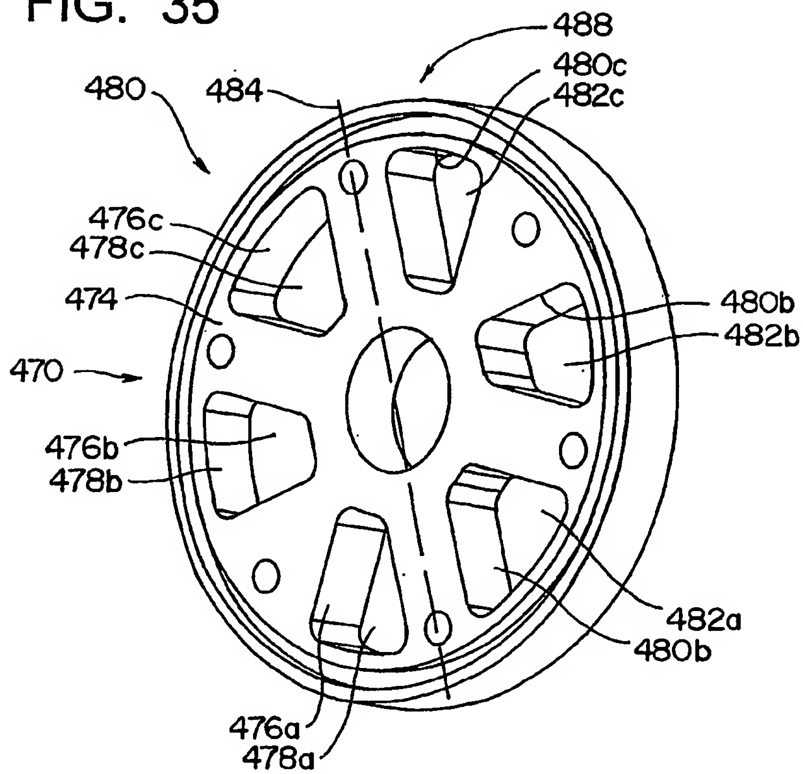

224a
453
450
430e
234e
232c
231c
230c
452
240
455
470
382
232d
394
398
305
231d
382
502c
502b
454
307
500d
230d
500c
254d
258d
500b
502a
234d
500a
430d

FIG. 35



488
480c
482c
480
484
476c
478c
474
470
480b
482b
476b
478b
482a
480b
476a
478a

FIG. 36A

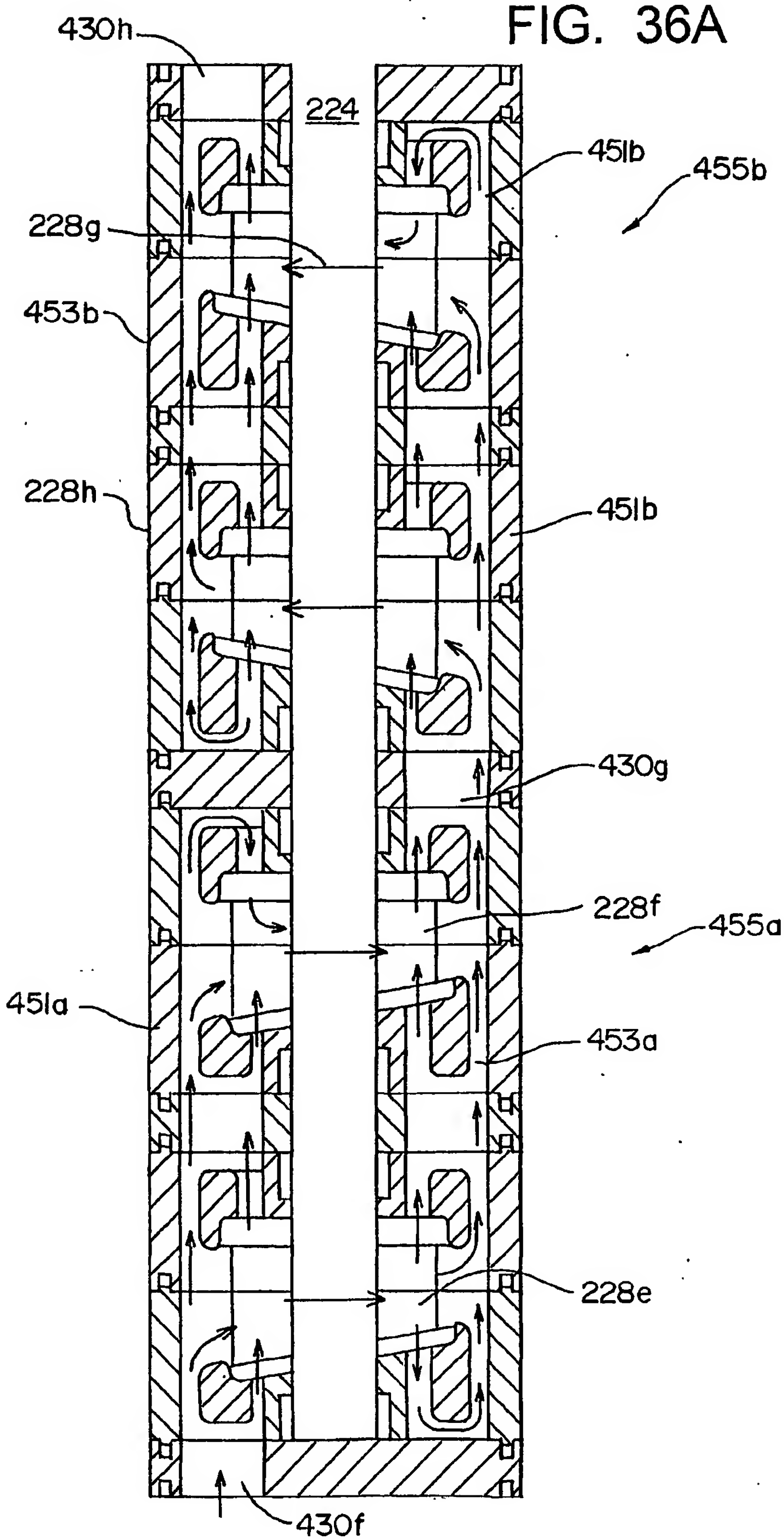

430h
224
451b
455b
228g
453b
228h
451b
430g
228f
455a
451a
453a
228e
430f

FIG. 36B

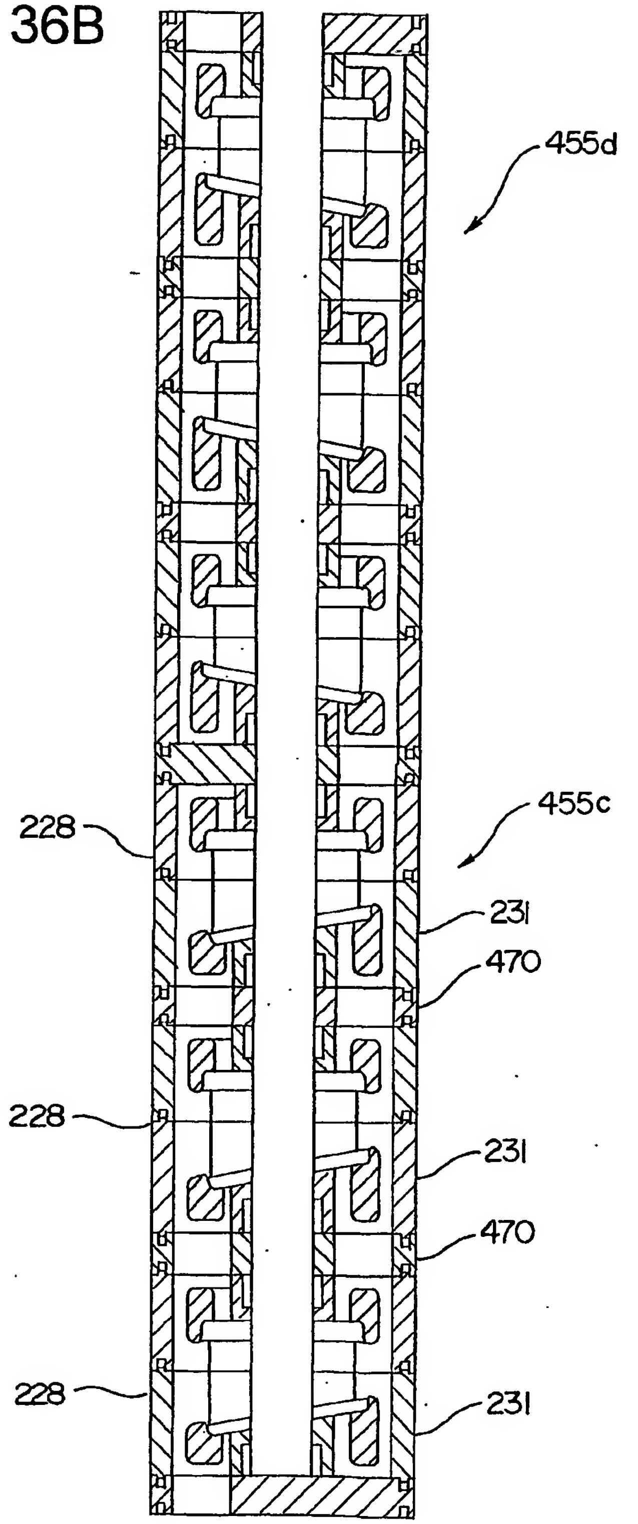


FIG. 37

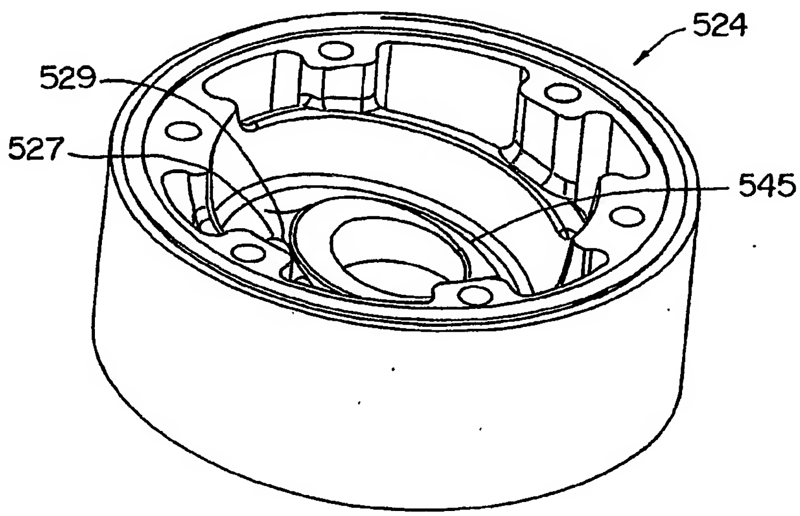


FIG. 38

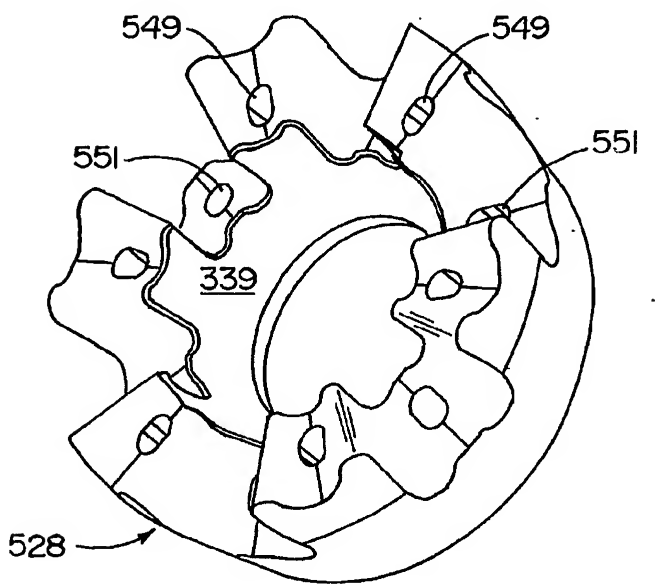


FIG. 39

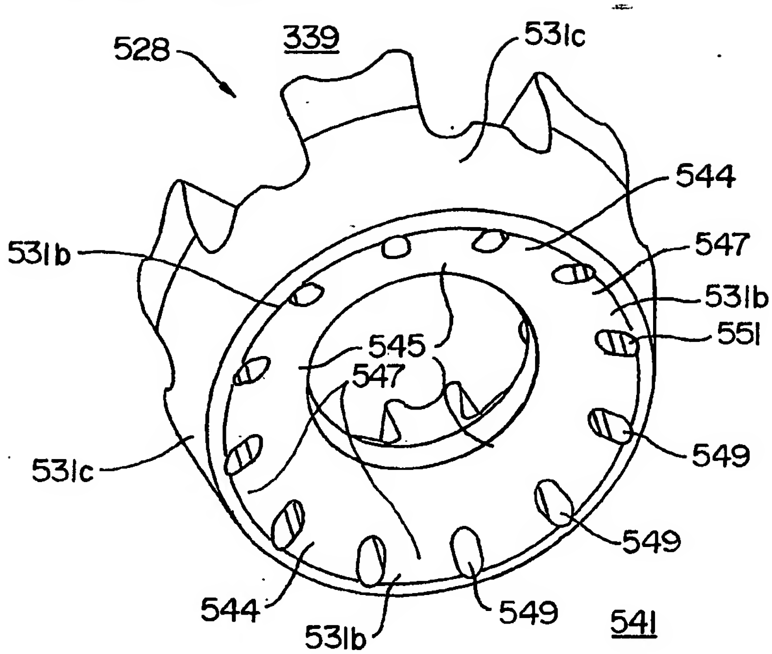

528
339
531c
544
547
531b
531b
551
545
547
549
531c
549
544
531b
549
541

FIG. 40

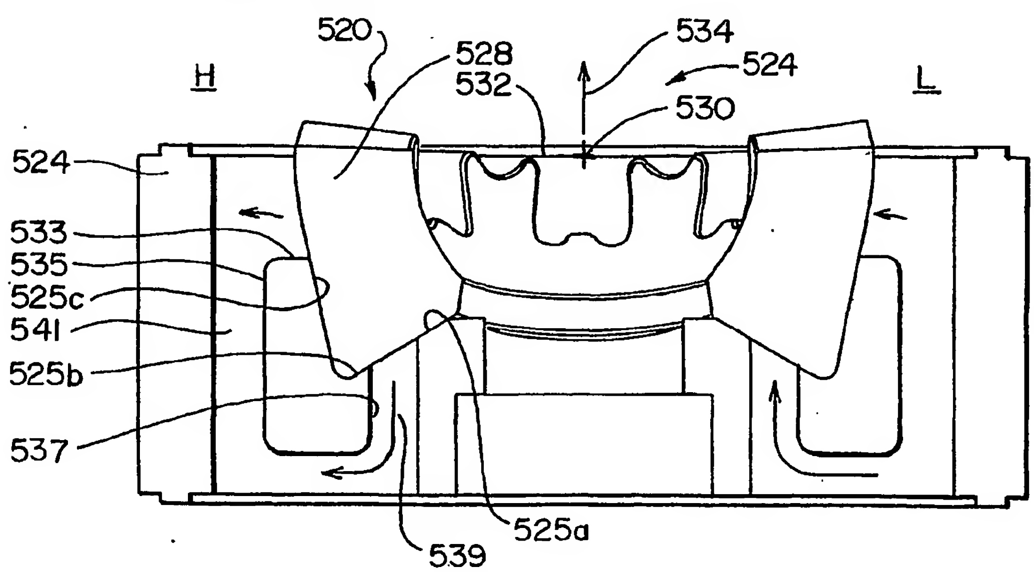

520
528
532
534
524
530
H
L
524
533
535
525c
541
525b
537
539
525a

FIG. 41A

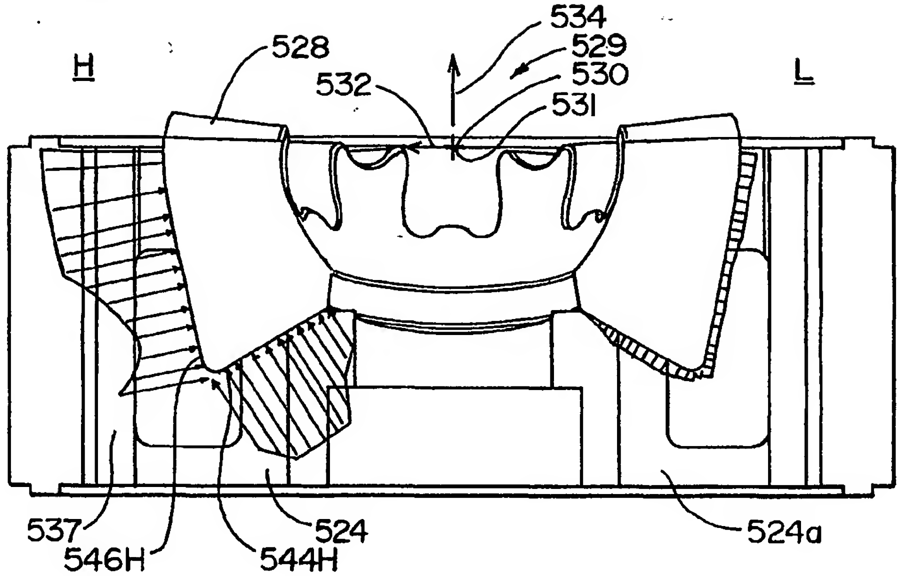


FIG. 41B

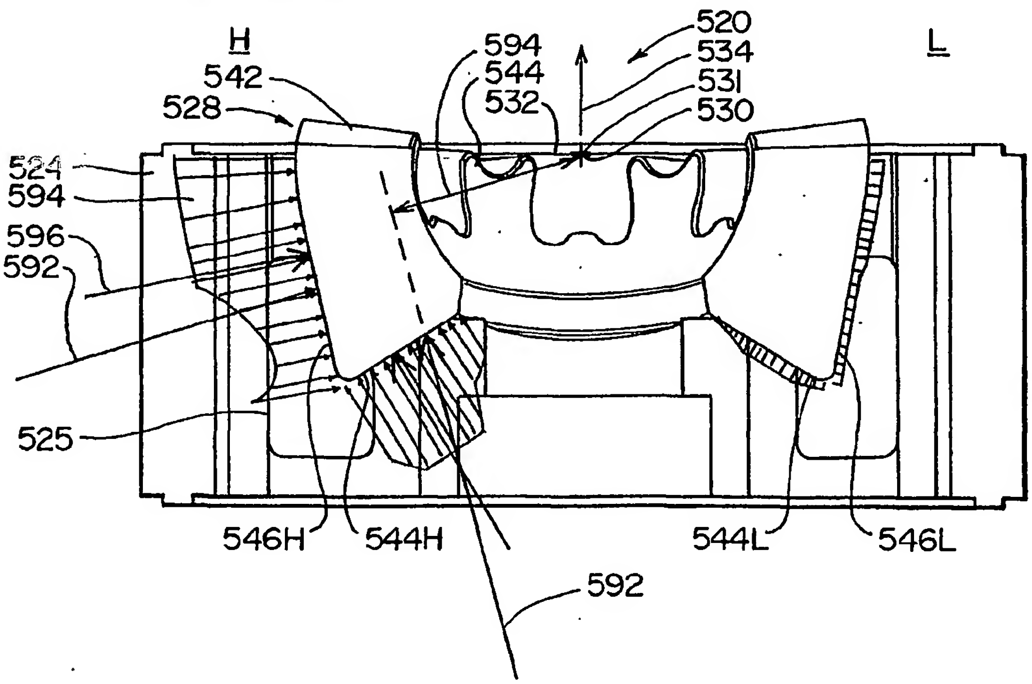


FIG. 42

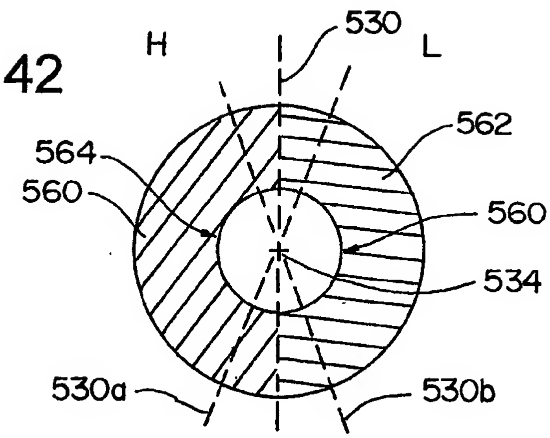


FIG. 43

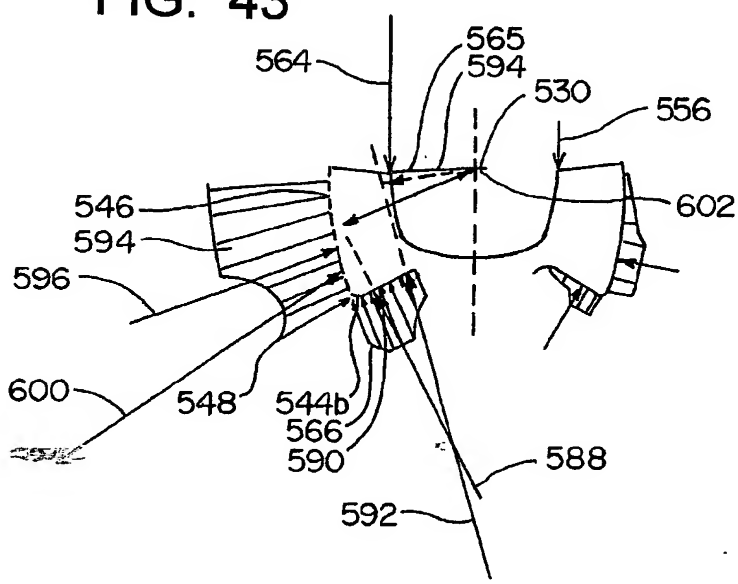


FIG. 45

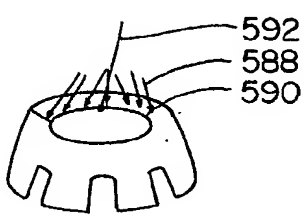


FIG. 46

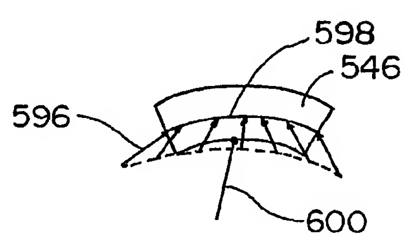


FIG. 44

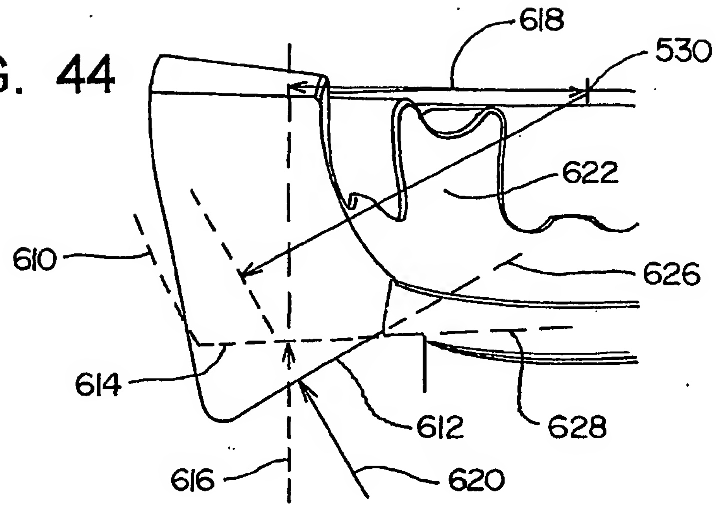


FIG. 44B

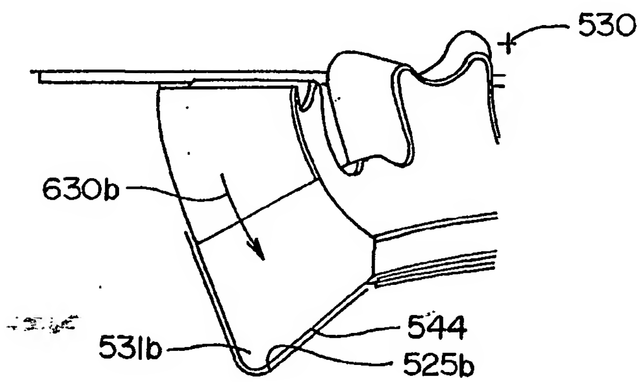


FIG. 44A

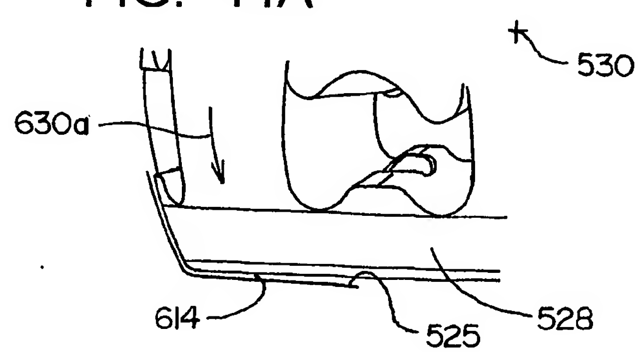


FIG. 47

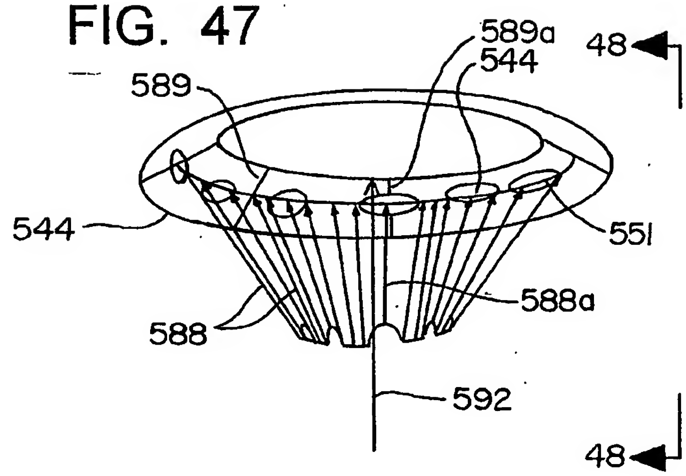


FIG. 48

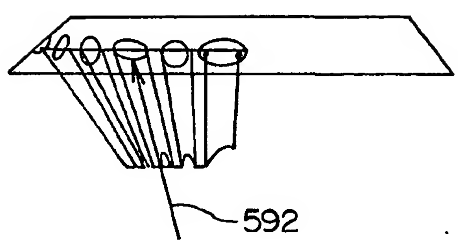


FIG. 49

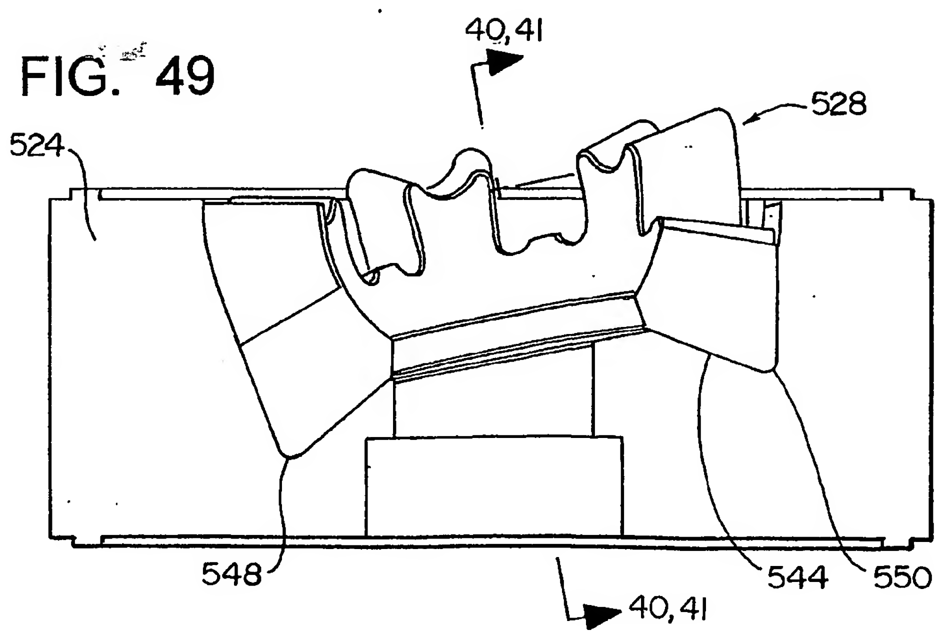


FIG. 50A

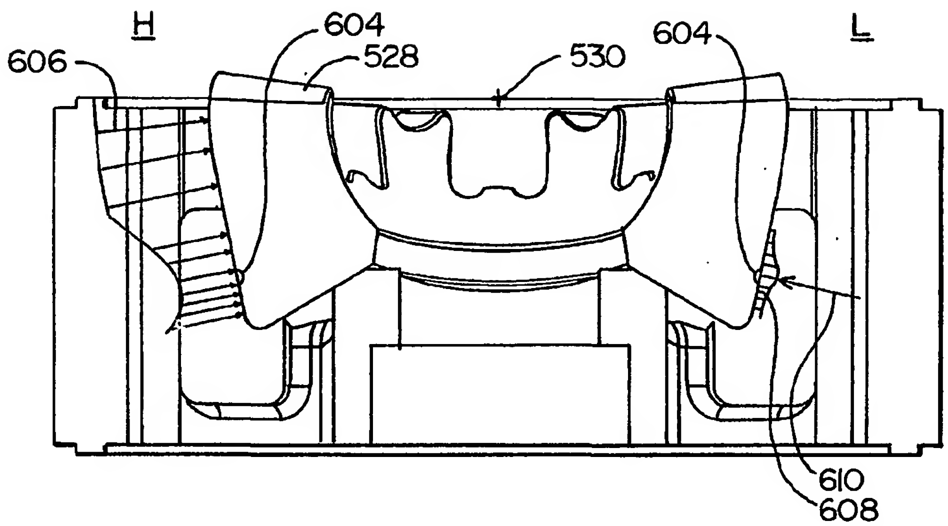


FIG. 51

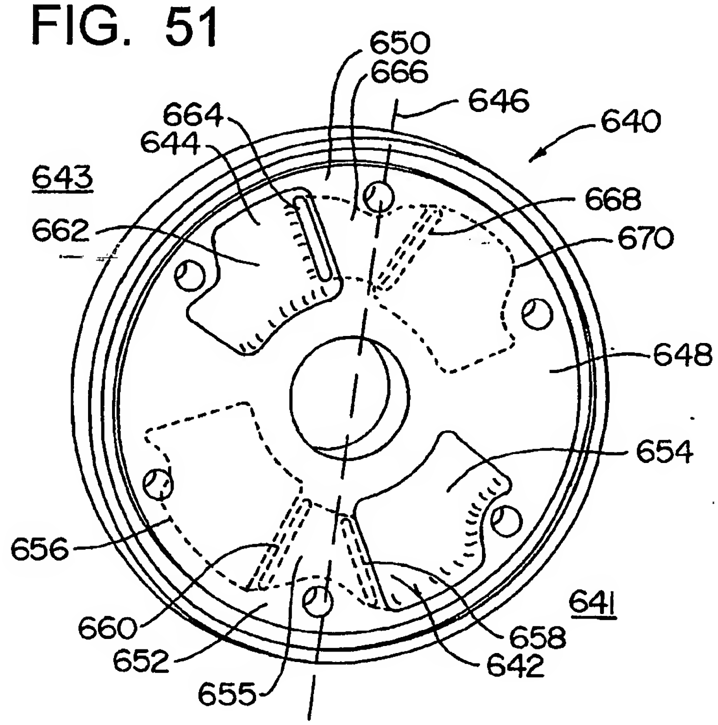


FIG. 50B

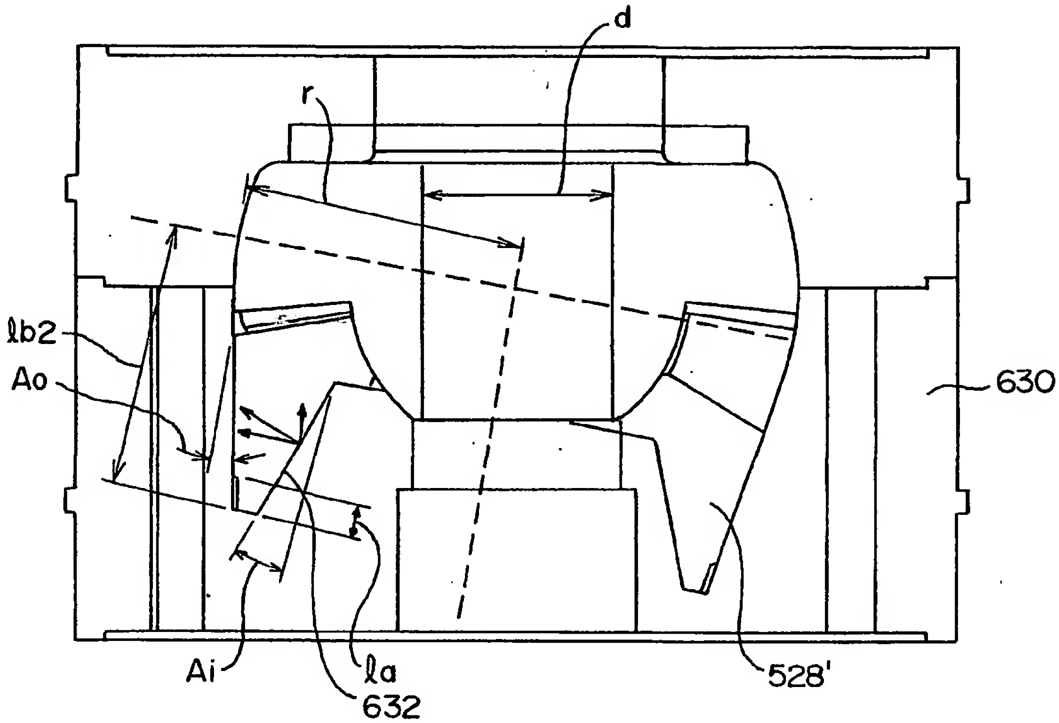


FIG. 52

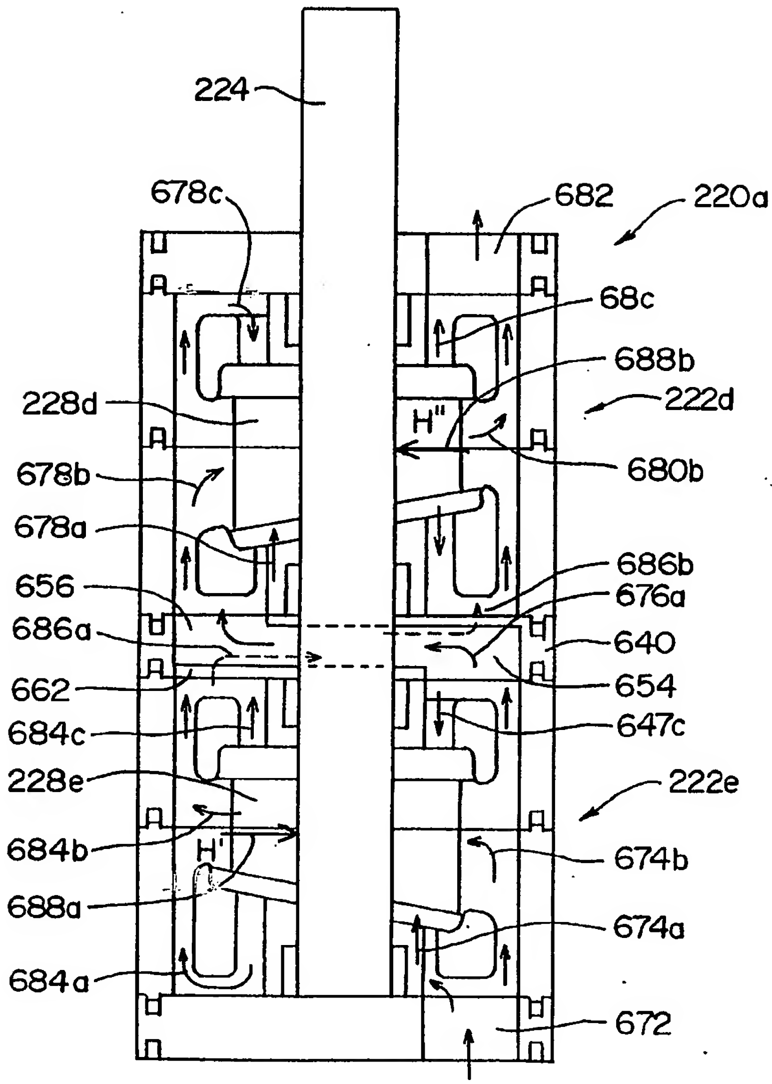

224
678c
682
220a
68c
688b
228d
222d
H"
680b
678b
678a
656
686b
676a
686a
640
654
662
684c
647c
228e
222e
684b
H'
674b
688a
674a
684a
672

FIG. 53

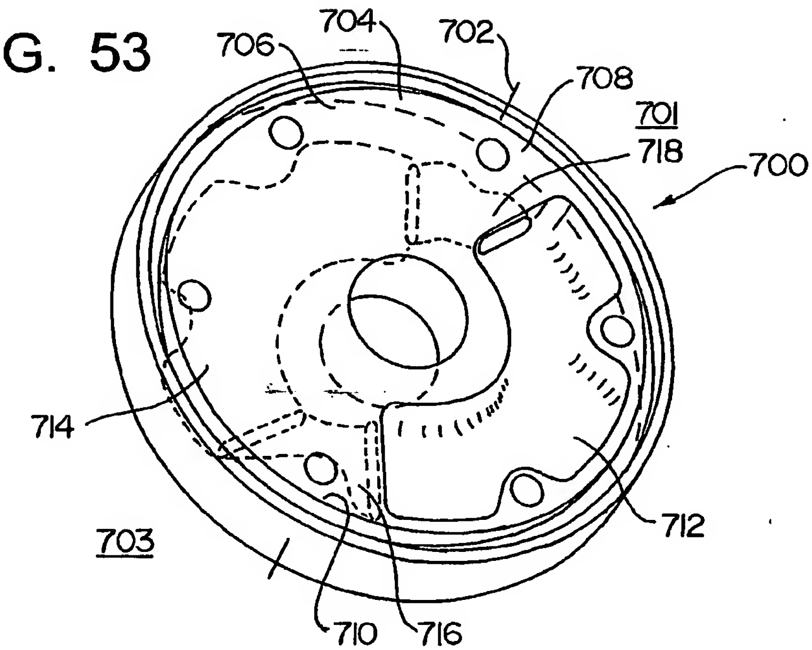

704
706
702
708
701
718
700
714
703
712
710
716

FIG. 54

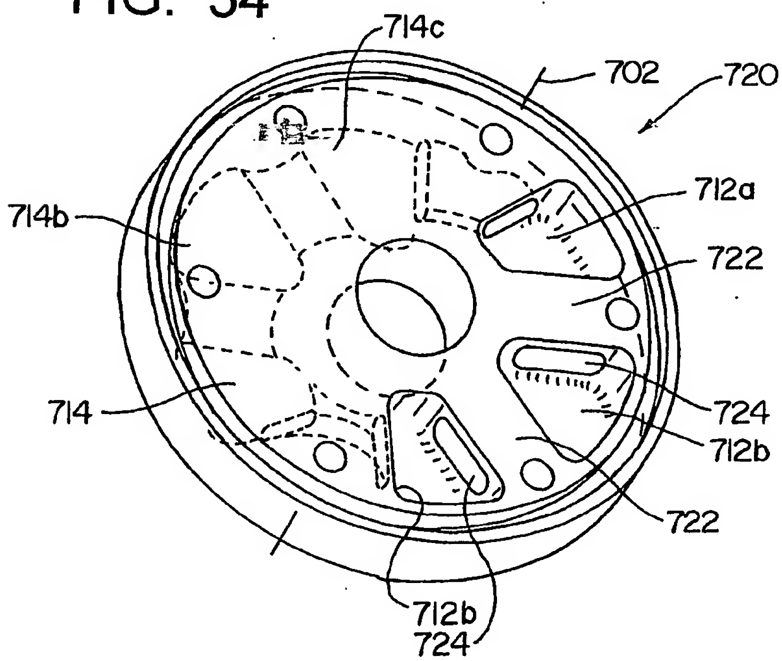

714c
702
720
714b
712a
722
724
714
712b
722
712b
724

FIG. 55

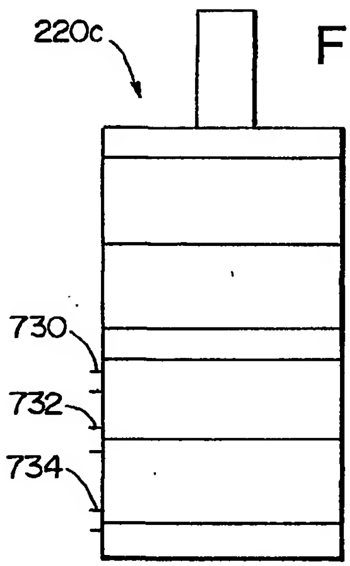

220c
730
732
734

FIG. 56

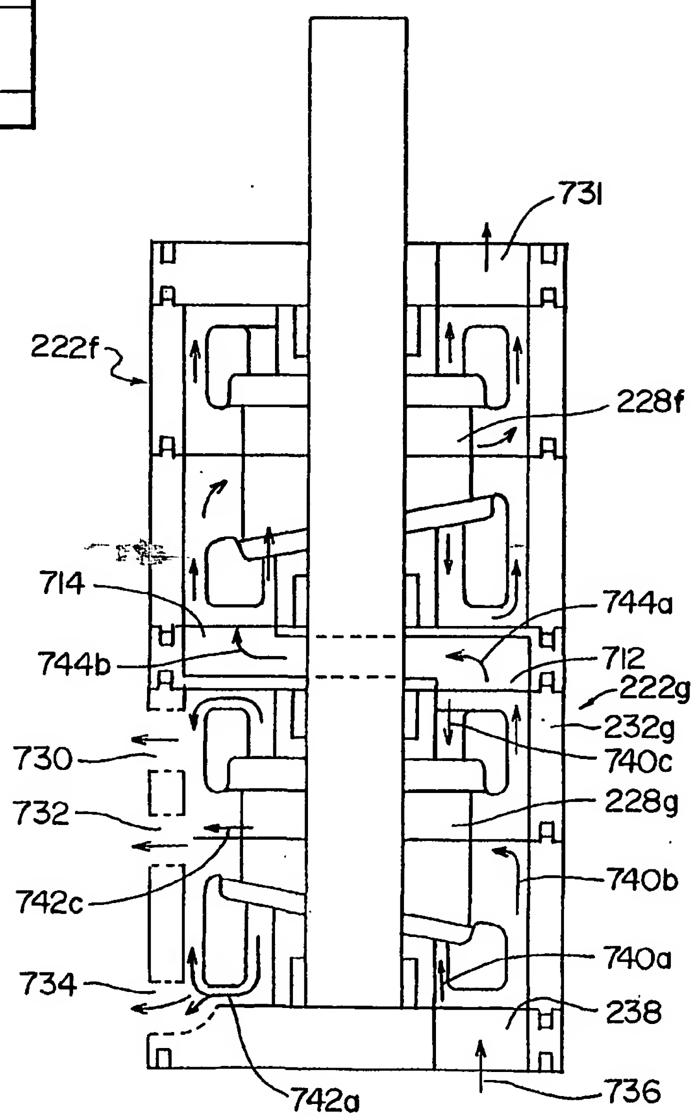

731
222f
228f
714
744a
744b
712
222g
232g
730
740c
228g
732
740b
742c
740a
734
238
742a
736

FIG. 57

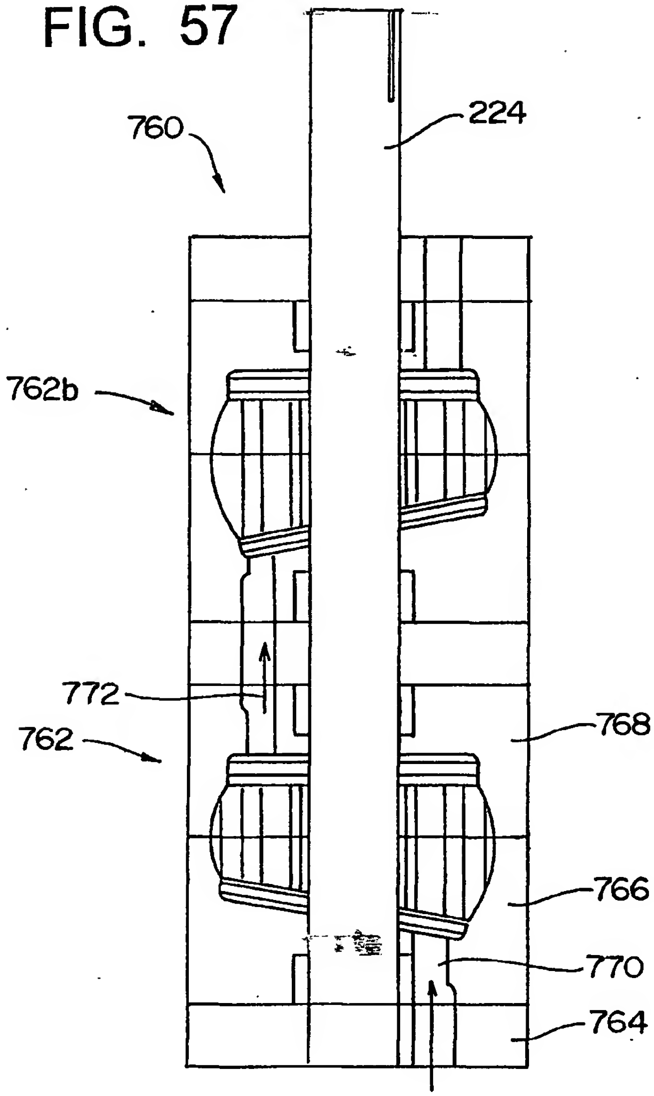
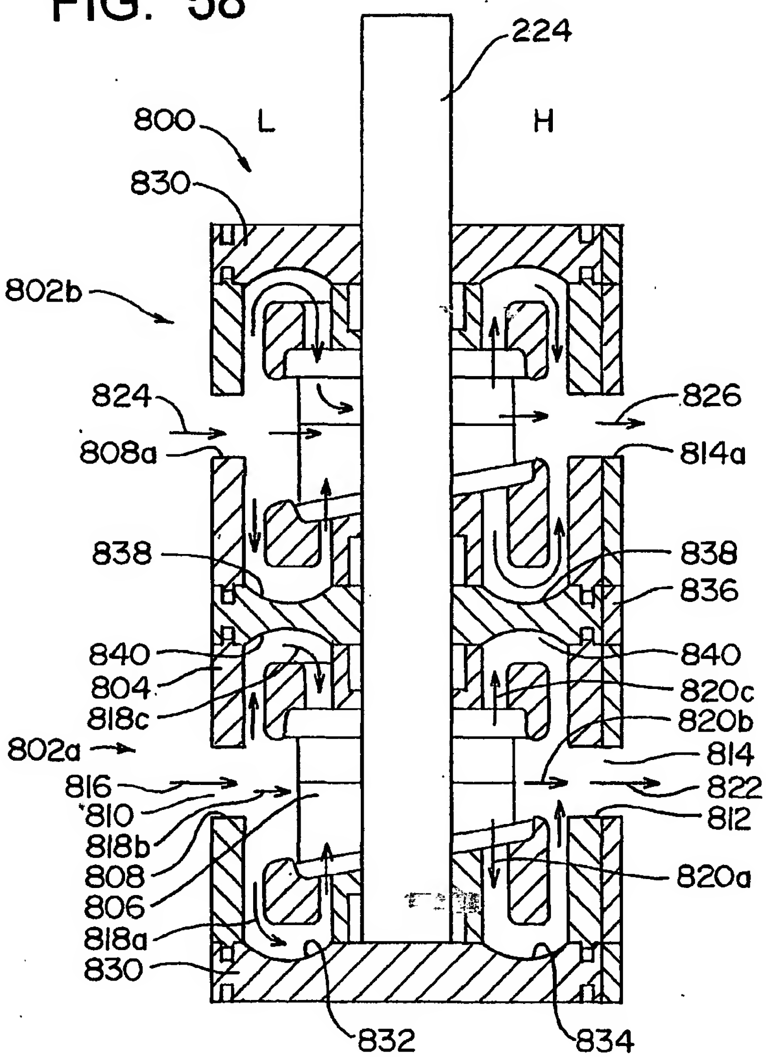


FIG. 58



224
800
L
H
830
802b
824
826
808a
814a
838
838
836
840
840
804
820c
818c
820b
802a
814
816
822
810
812
818b
808
820a
806
818a
830
832
834

FIG. 59

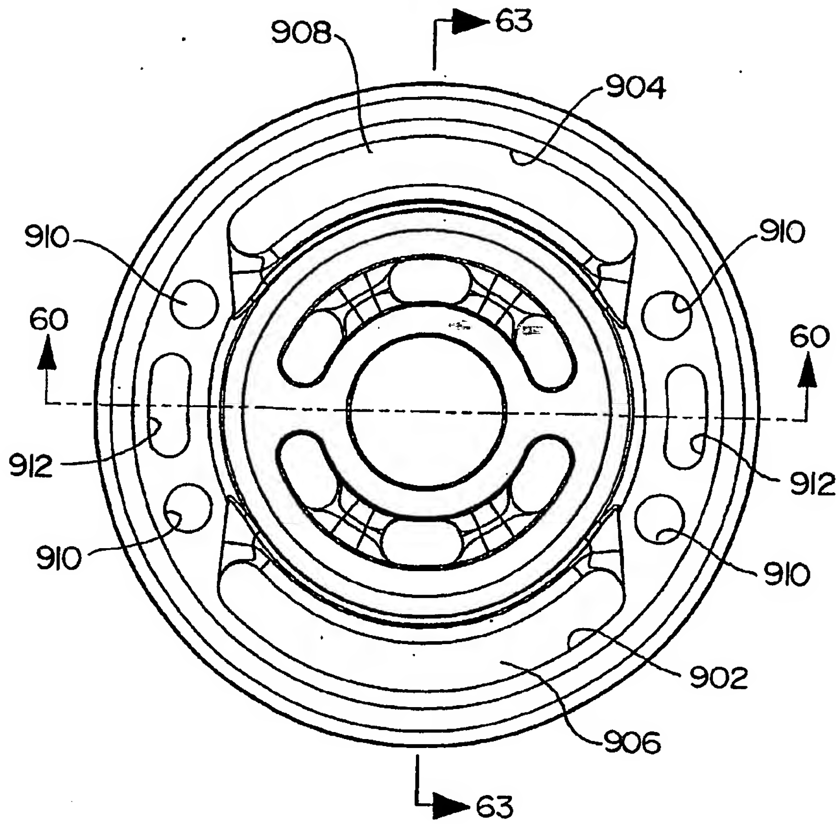


FIG. 60

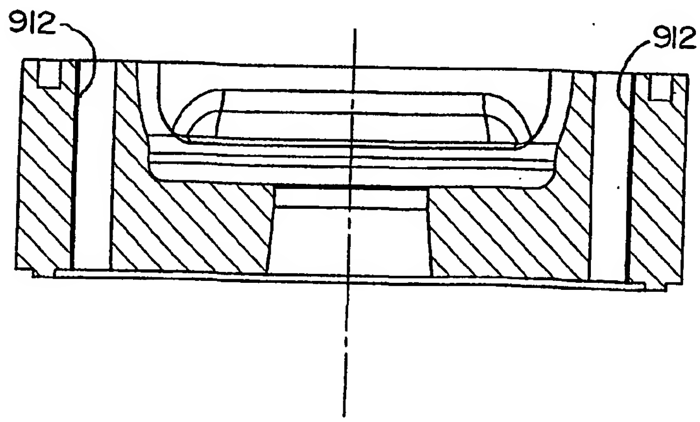


FIG. 61

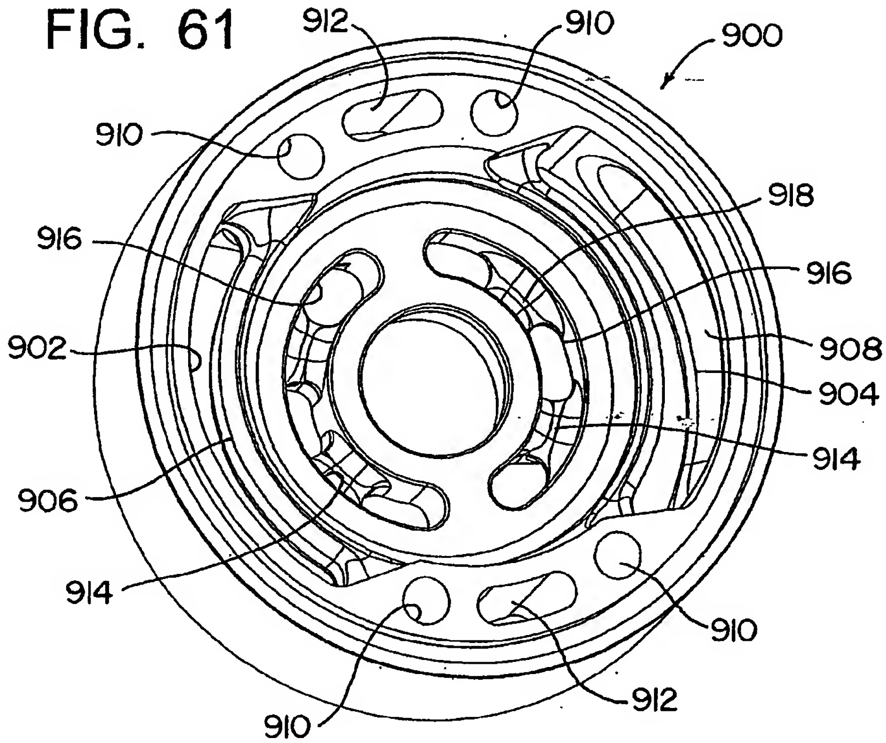

912
910
900
910
918
916
916
902
908
904
914
906
914
910
912
910

FIG. 62

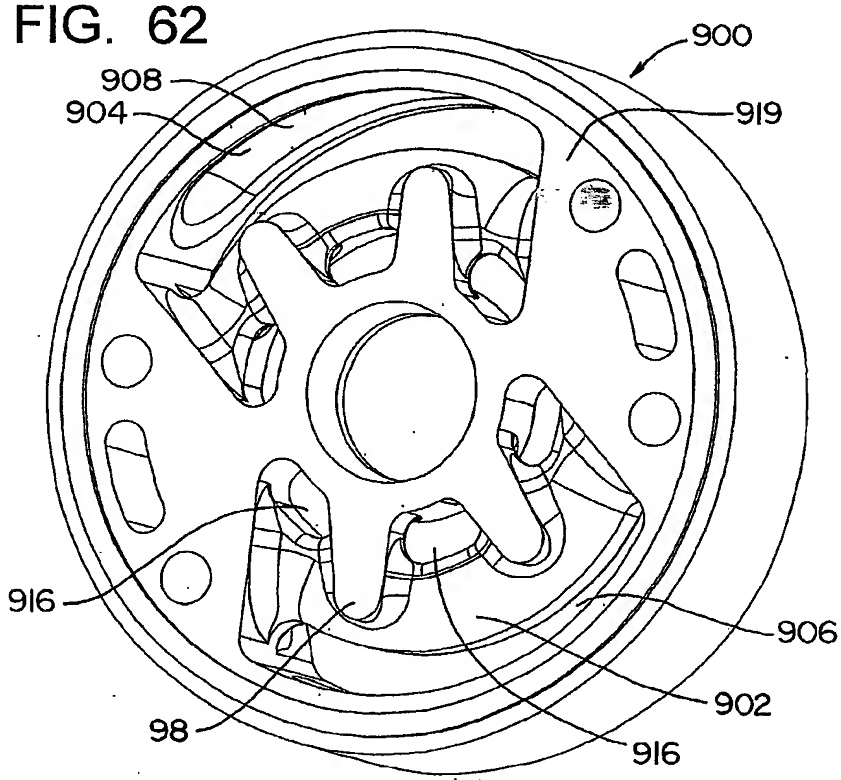

900
908
904
919
916
906
902
98
916

FIG. 62A

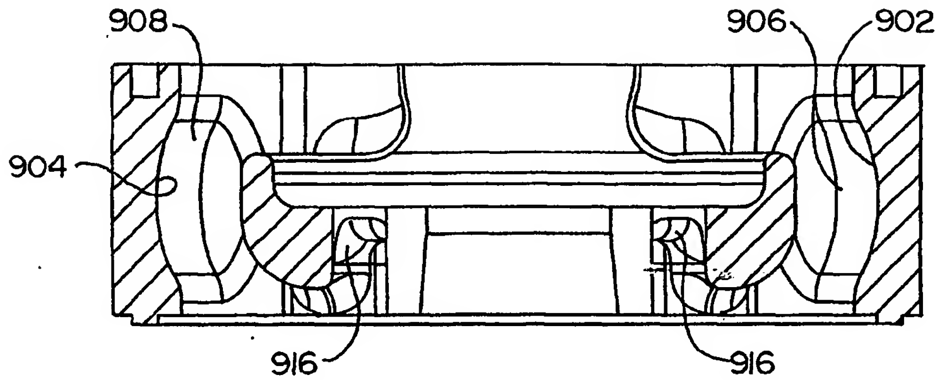


FIG. 63

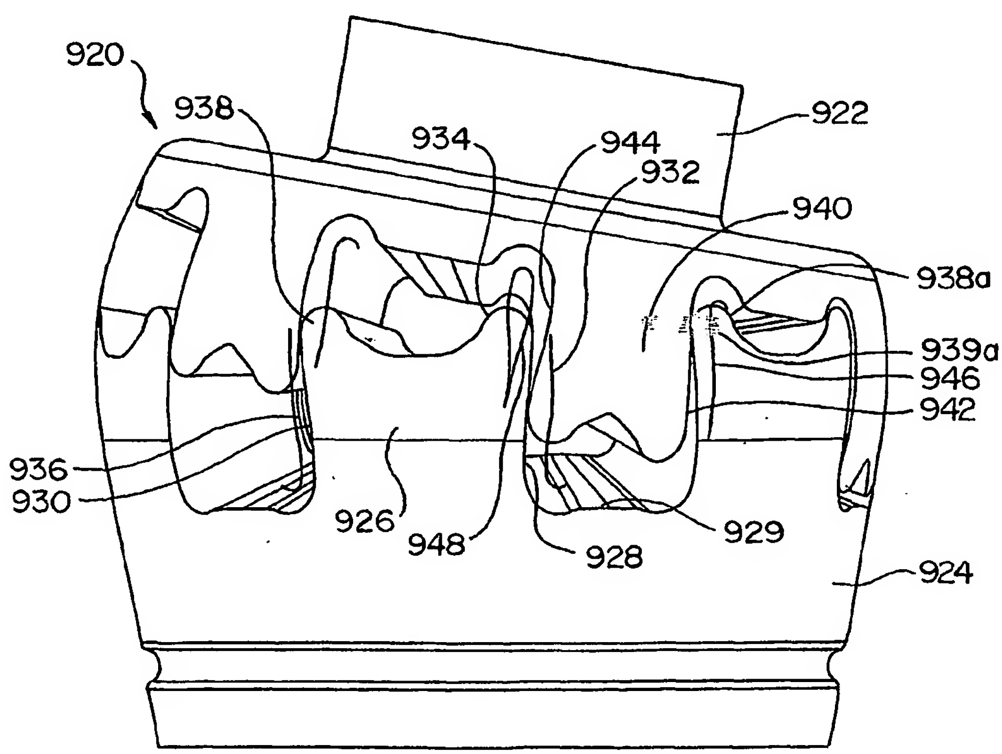


FIG. 64

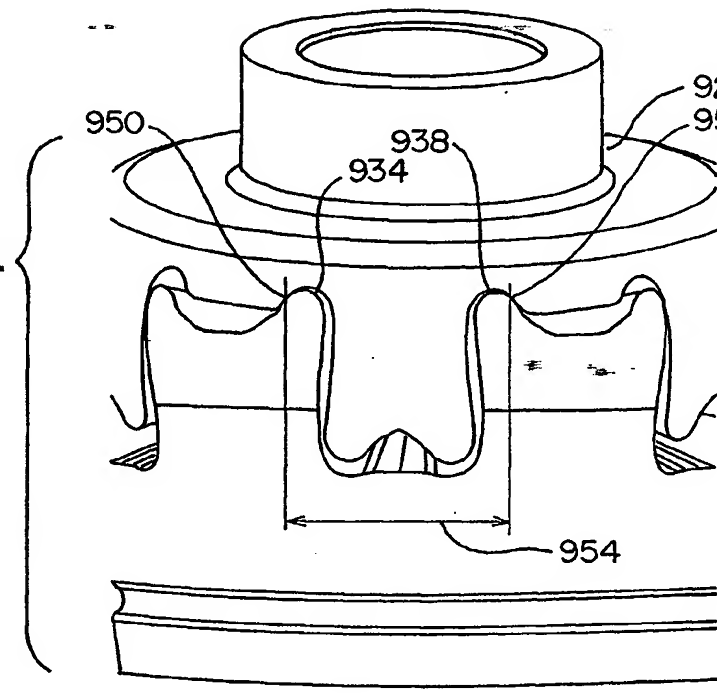
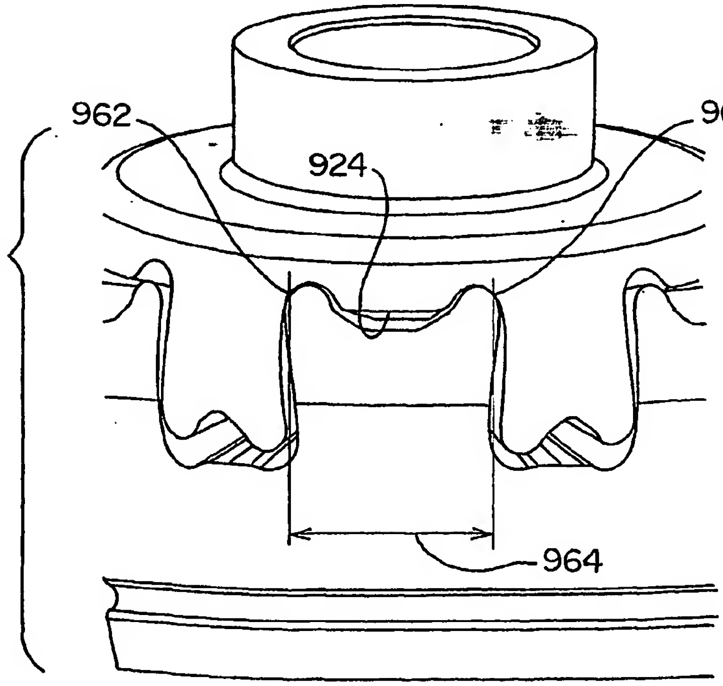


FIG. 65



INTERNATIONAL SEARCH REPORT	International application No. PCT/US01/22394

A. CLASSIFICATION OF SUBJECT MATTER

IPC(7) : F01C 3/08, 11/00; F03C 2/08

US CL : 418/9, 183, 186,195, 210, 212

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

U.S. : 418/9, 183, 186,195, 210, 212

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X --- Y	US 3,101,700 A (BOWDISH) 27 August 1963 (27.08.1963), column 4, line 75 to column 5, line 21.	1-4,25-28,31-33,46-50,61-64,68,69 --- 5-24,34-41,43,51-60,65-67
X --- Y	DE 3221994 A (WERNER) 15 December 1983 (15.12.1983), abstract and figure 8.	1-4,25-29,31-33,46-50,61-64,68,69 --- 5-24,30,34-45,51-60,65-67,70-75
Y	GB 5686 A (WEISS) 27 November 1902 (27.11.1902), page 1, lines 23-33.	5-24,35-45,51-60
Y	US 2,431,817 A (MANN) 02 December 1947 (02.12.1947), column 5, lines 3-28.	30,38-45,70-75
Y	US 914,155 A (MILLS et al) 02 March 1909 (02.03.1909), page 12, line 55-76.	35,37,66,67

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Date of the actual completion of the international search	Date of mailing of the international search report
05 September 2001 (05.09.2001)	02 OCT 2001
Name and mailing address of the ISA/US Commissioner of Patents and Trademarks Box PCT Washington, D.C. 20231 Facsimile No. (703)305-3230	Authorized officer T. Hurley for John J. Vrablik Telephone No. (703) 308-0861

INTERNATIONAL SEARCH REPORT

International application No.
PCT/US01/22394

C. (Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	U 3,272,130 A (MOSBACHER) 13 September 1966 (13.09.1966), colunm 2, line 39-51.	35,65

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